YACHT ARCHITECTURE:

A TREATISE

ON

THE LAWS WHICH GOVERN THE RESISTANCE OF BODIES MOVING IN WATER; PROPULSION BY STEAM AND SAIL; YACHT DESIGNING; AND YACHT BUILDING.

BY

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PREFACE.

In publishing the first edition of "Yacht Architecture" in 1885 I was able to state that the knowledge of the scientific principle on which yacht designing is based had been considerably extended since the publication of my large work entitled "Yacht Designing" in 1876. This can also be said of the period between 1885 and 1897, and no doubt the present generation of yachtsmen and those interested in yachts have a much more exact knowledge of the science of Naval Architecture than their predecessors had, owing to their having studied the numerous works published on the subject.

It cannot be claimed that any great discovery has been made since 1887 which has thrown new light on the science; notwithstanding yacht designing and constructing have undergone great development in all their branches. Nevertheless, the rescinding of the old tonnage rule as a means for rating yachts in competitive sailing, and, as a consequence, the removal of the tax upon beam has had a great effect upon the form of yachts, and enabled designers to solve the problems set them with greater freedom.

To meet the progress made in the science of designing and constructing steam and sailing yachts, the text has been thoroughly revised, and in many instances entirely rewritten; whilst a number of new plates of the lines of the most modern yachts take the place of those published in former editions. For most of these plates I am indebted to the liberality of the designers, who, I am proud to say, recognise what an advantage it is for all concerned that an exact knowledge of such an interesting science as that of yacht designing should be disseminated. In particular I am indebted to Mr. G. L. Watson; Mr. William Fife, jun.; Mr. C. P. Clayton; Mr. A. E. Payne; Mr. J. M. Soper; Mr. C. Sibbick; Mr. Linton Hope, &c.

Since 1876 much additional light has been thrown upon many hitherto obscure points in the laws which govern resistance by the further researches of the late Mr. W. Froude, and those of his son

Mr. R. E. Froude, who succeeded him as chief of the Admiralty Experimental Works. To both of these gentlemen I am largely indebted for the liberal manner in which they responded to my requests for information upon any points which seemed involved in obscurity; and, by the aid of their researches, the effect of form on resistance is now perhaps as well defined in these pages as it could be in the limited space at command. Some actual results of experiments made by the late Mr. W. Froude on forms for gunboats are given; not because anyone will be likely to build such vessels for pleasure, but for the reason that these particular experiments throw light upon the whole nature of the resistance of bodies moving in water.

A large space is devoted to steam yachting, and it should be mentioned that Chapter XI., which deals with the boiler and engine, was mainly written by Mr. G. R. Dunell, whose practical knowledge of the subject and literary ability rendered his assistance of the greatest value. In writing Chapter XII., on Steam Propulsion, I had to apply to many builders and engineers for particulars as to trial trips and performances, and in all cases I met with a ready response, and as much is said about the subject as the owner of a steam yacht is likely to desire to know.

The system of designing provided is practically identical with that published in "Yacht Designing" in 1876, but revised and extended where necessary. As explained in that work, it is not intended to be arbitrarily followed, and was prepared as a guide for those whose experience may be too limited to much assist their judgment in making a design. The system has been tried for twenty years, and has been found to answer its purpose well; and this much can be said for it, that if a strikingly good vessel could not be designod by rigidly following the method, a very bad one would not be the result.

Yacht building can be better described by engravings than by writing, and the plates and woodcuts which are given will instruct the amateur in all the details of construction according to the practice of the best builders.

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For the year 1897, and until further change, the Yacht Racing Association has decided that centre-boards of all kinds shall be measured as fixed keels, instead of in special cases, at 1·5 times their extreme drop.

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Series A.—Designs of Yachts.

Series B.

Series C.—Steam Yachts.

Series D.—Boats.

ERRATA.

PAGE 104.—Sixth line from top, *read* Ailsa *for* Ceilsa.

PAGE 275.—Third line from bottom, *for* $\frac{i\,e}{i\,a} = \theta$, *read* $\frac{i\,e}{i\,a}$ = tangent θ.

PAGE 280.—First line, *for* $\left(\frac{10,133 \times S}{P \times 100 - k}\right)$, *read* $\left(\frac{10,133 \times S}{P \times (100 - k)}\right)$.

PLATE XII.—KISMET.—This yacht was built by the Thames Yacht Building Co., Greenhithe, Kent, for Mr. Linton Hope, from his own design.

PLATE XXV.—The sail area of Induna is 260 square feet, not 200 square feet.

YACHT ARCHITECTURE.

CHAPTER I.

DISPLACEMENT AND BUOYANCY.

THE displacement of a vessel is the quantity or bulk of water (generally represented by a measure or weight) which a vessel displaces or pushes away when she is put into the water. This quantity of water is always equal to the whole weight of the vessel and everything that she contains; that is to say, the vessel will sink into the water until she has displaced or pushed away a quantity of the fluid equal to her own weight and the weight of everything that she contains, and then will sink no further.

If the weight of water displaced is also exactly equal in *bulk* to the *bulk* of the vessel, then the vessel will sink in the fluid until her entire bulk is immersed; or, in other words, if the body immersed be a solid of the specific gravity of the water, then will the solid sink into the fluid until it is entirely immersed. For example, a cubic foot of African oak weighs 62lb., a cubic foot of fresh water weighs 62½lb., and consequently, if a cubic foot of African oak were placed in fresh water, it would nearly sink to the level of the surface; but a cubic foot of *sea water* weighs 64lb., and, consequently, if a cubic foot of African oak were placed into sea water, it would sink until 62lb. of the fluid were displaced (which would be less than a cubic foot), and would sink no deeper, so practically 2lb. of the oak cube would remain above the surface, and this would be termed surplus buoyancy.

This well illustrates the meaning of the terms "displacement" and "surplus buoyancy." A vessel weighs, we will say, with all her ballast, spars, sails, gear, stores, crew, and everything belonging to her on board, one ton; then, if she is put into the water, she will *displace* exactly one ton of the fluid. Now a ton of sea water in bulk contains 35 cubic feet; consequently, if the *bulk* of the vessel only equalled 35 cubic feet, she would sink into the water until entirely immersed. But a vessel that weighed one ton would contain in *bulk* a great deal more than 35 cubic feet, measuring her actual

body on the outside from keel to deck as if she were a solid; that is, the whole body or bulk of the vessel so measured would probably equal 60 cubic feet. The result would be that the vessel would sink into the water until 35 cubic feet of the hull became immersed, and sink no further; this would leave 25 cubic feet above the water.

The buoyancy of a vessel may be taken as a force equal to the weight of water it displaces; or, in other words, any given weight of fluid will support a similar solid weight of equal bulk. The quantity, or bulk, of a fluid which a vessel will displace depends on the density of that fluid, as previously explained. Thus sea water is denser, or more buoyant, than fresh water: and, consequently, a cubic foot of sea-water will support a greater weight in the same bulk than a like quantity of fresh water. Mercury is a fluid so dense that even iron will float in it with only a little more than half its bulk immersed, for the reason that a cubic foot of mercury weighs 849lb., whereas a similar bulk of iron only weighs 480lb.

Thus the displacement of a vessel is always equal to her own weight, including the weight of everything and every person on board; and providing that the bulk or size of the body of water displaced is smaller than the bulk or size of the vessel (regard her from deck to keel), then a portion of the vessel will always be above the surface of the water, and this portion of a vessel is called her freeboard, and spare or surplus buoyancy.

The truth of the foregoing can be demonstrated by a simple experiment. Take a large basin, such as A (Fig. 1), and fill it carefully to the

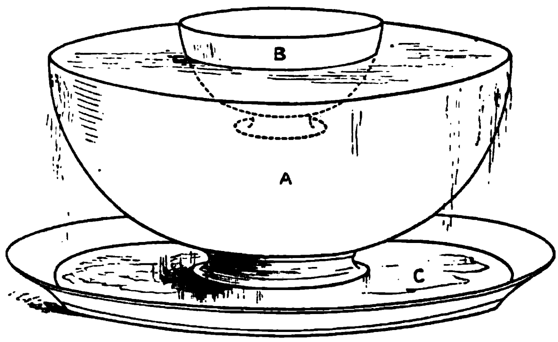


FIG. 1.

brim with water, and stand it in the saucer, C. Then take a smaller basin, B, and put it into the water, which of course will overflow into the saucer. If the water that so overflows and the small basin be afterwards put into a scale and separately weighed, their weight will be found to be exactly equal; and, further, if shot or other substance be put into the small basin B whilst it is floating, still more water will overflow, and if the whole

of the water which so overflows be weighed, and the small basin and its contents be weighed, their respective weights will be proved equal.

This experiment can be utilised to arrive at the displacement of the ship from that of the model. Thus, say a model of the Kriemhilda* is made to half an inch scale (or one twenty-fourth of her real dimensions), and put into a trough filled with salt water to the aperture of a waste pipe; then as the model became immersed, the water would escape by the waste pipe into some vessel, say a large bucket. The escaped water in this particular case would weigh 18·69lb. Now the displacement of the ship to that of the model is simply as the cube of the difference in the dimensions; or say the scale for the ship is twelve times greater than that for the model; then the displacement or weight of the model multiplied by the cube of 12 ($12^3 = 1728$) would give the displacement of the ship. In the case given above, the real yacht was to be 24 times greater than that of the model which was on a half inch scale, and the weight of which was 18·69lb.; the cube of 24 is 13,824, and 18·69lb. multiplied by 13,824 is equal to 258,370lb. There are 2240lb. to one ton; then 258,370 divided by 2240 gives a quotient of 115·33 tons, the exact displacement of Kriemhilda. If the model is made to other scales the displacement can be found by a similar process.

If the model of the yacht be accurately made from the drawings, this experiment will give the exact displacement of a yacht when afloat and ballasted. (Examples of calculating the displacement from the drawings will be given further on).

Scale of model to 1ft. INCH.	Proportion to real ship.	Cube of proportion by which the weight in lb. of displacement of the model is multiplied.	Multipliers.
1	12	1728	0·771
$\frac{7}{8}$	13·7143	2579	1·051
$\frac{3}{4}$	16	4096	1·828
$\frac{5}{8}$	19·2	7077·9	3·160
$\frac{1}{2}$	24	13824	6·171
$\frac{3}{8}$	32	32768	14·629
$\frac{1}{4}$	48	110592	49·371
$\frac{1}{8}$	96	884736	394·971

The weight of the model in pounds can be converted into tons by using the multipliers set out in the last column in the table given above: thus, take the weight of Kriemhilda's model (to half inch scale), and multiplying by the factor 6·171 we have 18·69lb. × 6·171 = 115·33 tons.

On a similar principle, if a drawing is made, say to a $\frac{3}{8}$in. scale, and it should be desired to see what the displacement would be by a $\frac{1}{4}$in. scale, the displacement by the $\frac{1}{4}$in. scale would have to be multiplied by the cube of the proportion of one scale to another. The $\frac{3}{8}$in. scale would be $\frac{1}{32}$ of

* The calculation of this yacht's displacement illustrates the text farther on.

the real ship, and the ¼in. scale $\frac{1}{48}$; the proportion of one scale to the other would therefore be $\frac{48}{32}$ = 1·5. The cube of 1·5 is 3·375, and the displacement by the ⅜in. scale would have to be multiplied by that factor. Thus, say the displacement is 12 tons, 12 × 3·375 = 40·5 tons = the displacement by the ¼in. scale. Similarly, if a ⅜in. scale has to be applied to a ¼in. drawing the process would be the same, except that the displacement by the ¼in. scale would have to be divided by the cube of the proportion of one scale to the other. In the case just treated we should therefore have (proportion = 1·5; and $1·5^3$ = 3.375). $\frac{40·5 \text{ tons}}{3·375}$ = 12 tons.

The following table of factors or divisors may be useful.

⅛in. to	Proportion.	Cube of Proportion. Factor, or Divisor.	¼in. to	Proportion.	Cube of Proportion. Factor, or Divisor.	⅜in. to	Proportion.	Cube of Proportion. Factor, or Divisor.	½in. to	Proportion.	Cube of Proportion. Factor, or Divisor.
¼in.	2·000	8·000	⅛in.	2·000	8·000	⅛in.	3·000	27·000	⅛in.	4·000	64·000
⅜in.	3·000	27·000	⅜in.	1·500	3·375	¼in.	1·500	3·375	¼in.	2·00	8·000
½in.	4·000	64·000	½in.	2·000	8·000	½in.	1·333	2·369	⅜in.	1·333	2·369
⅝in.	5·000	125·000	⅝in.	2·500	13·824	⅝in.	1·666	4·624	⅝in.	1·250	1·953
¾in.	6·000	216·000	¾in.	3·000	27·000	¾in.	2·000	8·000	¾in.	1·500	3·375
⅞in.	7·000	343·000	⅞in.	3·500	42·875	⅞in.	2·333	12·689	⅞in.	1·750	5·359
1in.	8·000	512·000	1in.	4·000	64·000	1in.	2·666	18·949	1in.	2·000	8·000

⅝in. to	Proportion.	Cube of Proportion. Factor, or Divisor.	¾in. to	Proportion.	Cube of Proportion, Factor, or Divisor.	⅞in. to	Proportion.	Cube of Proportion, Factor, or Divisor.	1in. to	Proportion.	Cube of Proportion. Factor, or Divisor.
⅛in.	5·000	125·000	⅛in.	6·000	216·000	⅛in.	7·000	343·000	⅛in.	8·000	512·000
¼in.	2·500	15·625	¼in.	3·000	27·000	¼in.	3·500	42·875	¼in.	4·000	64.000
⅜in.	1·666	4·624	⅜in.	2·000	8·000	⅜in.	2·333	12·698	⅜in.	2·666	18·949
½in.	1·250	1·953	½in.	1·500	3·375	½in.	1·750	5·359	½in.	2·000	8·000
¾in.	1·200	1·728	⅝in.	1·200	1·728	⅝in.	1·400	2·744	⅝in.	1·600	4·096
⅞in.	1·400	2·744	⅞in.	1·166	1·585	¾in.	1·166	1·585	¾in.	1·333	2·369
1in.	1·600	4·096	1in.	1·333	2·369	1in.	1·143	1·493	⅞in.	1·143	1·493

In yacht designing, the chief use of ascertaining the weight of the displacement is to determine the weight of ballast a sailing yacht will carry, and so be able thereby to form an approximate estimate of her stability.

In the case of steam yachts, it is of the greatest importance that the displacement should be calculated, in order that the yacht's capacity for carrying her machinery and coal may be estimated.

To ascertain the weight of ballast, the weight of hull or material used in the construction or fitting up a yacht must also be known, besides the weight of displacement; no fixed rule as to the proportion the weight of the ballast should bear to the whole weight of displacement, and the case of each yacht must be treated individually. (This matter will be discussed more fully under the head of "Ballast.")

Displacement is sometimes used as a test of the relative fullness or fineness of the immersed part of the hull by what is termed a co-efficient,

but the test is of little value, so far as yachts of the modern type are concerned. The co-efficient is the ratio the displacement bears to the circumscribing parallelopipedon. In Fig. 2, let *a b c d e f g h* represent a block out of which the hull has to be shaped; it is obvious that in carving away the hull to meet the form of the midship section at *i*, or the stem, keel, and sternpost at *j k l*, that the volume of the block will be considerably reduced, and the co-efficient will be found by dividing the

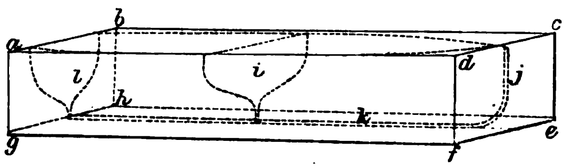


FIG. 2.

displacement of the model by the volume of the rectangular figure formed by the length, breadth, and *mean* draught of water. The depth from the load water-line to the rabbet of keel is sometimes used as the draught, but in the case of yachts it is preferable to take mean draught of water, as the rabbet is so variously placed with regard to the under water depth. The co-efficients of fineness of some well-known yachts of a bygone period are

	Thames Tons.	Co-efficient.	Displacement in Tons.
Gwendolin	197	·309	202·0
Jullanar	126	·359	158·0
Florinda	135	·311	150·0
Formosa	102	·318	130·0
Arrow	118	·262	106·0
Miranda	139	·310	160·0
Drina	13	·413	15·0
Sleuthhound	43	·331	63·0
Mascotte	3	·417	7·0
Vanessa	21	·305	28·5
Kriemhilda	106	·278	115·3
Vol-au-Vent	104	·281	116·0
Freda	5	·322	7·7
Genesta	80	·383	141·0
Beluga	56	·326	78·0
Ghost	23	·370	31·0
Saraband	12	·393	22·0
Neptune	11	·308	15·0
Oriental (S. Y.)	232	·460	266·0
Fair Geraldine (S. Y.)	301	·403	315·0
Marchesa (S. Y.)	160	·435	465·0
Chazalie (S. Y.)	545	·370	533·0
Jacamar (S. Y.)	451	·400	422·0
Bulldog (S. Y.)	60	·424	58·0
Celia (S. Y.)	27	·410	27·0
Linotte (S. Y.)	92	·403	84·0

The apparent differences in the relative fineness of these yachts is chiefly dependent on the form given to the outline of the vertical

longitudinal section, and a yacht with her forefoot very much cut up, such as Jullanar, Drina, Ghost, or with very little depth of keel below the rabbet, like a steam yacht, will appear to have a high co-efficient or great fulness, while older yachts, with a great deal of gripe or forefoot, and with considerable depth of keel, such as Arrow, Kriemhilda, Vol-au-Vent, &c., will appear to be unusually fine; but as a fact, there is not the variation in the relative fineness of the entrance and stern as the co-efficients would lead one to suppose. Therefore, unless the longitudinal-vertical outline of any yachts under comparison agree pretty closely, such as the outlines of most steam yachts do, the test of the fineness by a co-efficient may be very fallacious; and, of course, the test would be quite unreliable if applied to modern yachts of the fin keel type. A reliable test can, however, be applied by getting out a curve of displacement by vertical sectional areas. This matter will be considered when the form of the hull is reached.

It has been pointed out that a vessel will sink into the water until she has displaced a quantity of the fluid equal to her own weight: but there may still be a portion of the vessel unimmersed, and this portion would form what is termed the "surplus buoyancy." The value of this surplus buoyancy will be indicated in the chapter on stability, but there are no precise rules for determining the quantity that ought to exist in proportion to the under-water quantity. As a rule the bulk above water is nearly equal to the under-water bulk, but occasionally it is considerably less.

A rough rule for finding the cubic contents of the above-water bulk would be $L \times B \times H \times 0{\cdot}75$, where L length over all, B breadth, and H the least height of freeboard. (See also the chapter on "Dimensions.")

CENTRE OF BUOYANCY.

The centre of buoyancy of a vessel is the centre of the cavity or hole made in the water by the part of the vessel which is immersed; hence it is frequently termed the centre of gravity of the displacement. Through this centre the aggregate pressure of the surrounding fluid acts on the immersed part of the hull, and it is necessary to note that in speaking of the centre of buoyancy of a vessel, such as a yacht or a ship of any kind, only that portion which is immersed is necessarily considered, so that in fact the vessel is treated as if she were cut down to the surface of the water. The centre of buoyancy of an immersed plane of similar sides, Fig. 3, or of an immersed sphere, would necessarily be in the centre of the block at *o*—that is, would be equidistant from both sides and ends, and from top and bottom, and the centre of gravity of a rectangular solid (see Fig. 4) would be at the point where two lines *a b*, *c d* intersect each other at

e ; this is, however, assuming that the body is homogeneous, such as water, and it must not be forgotten that it is in reality the centre of the hole made

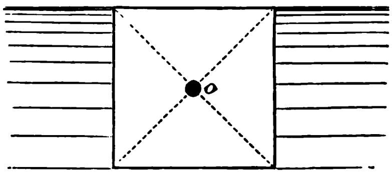


FIG. 3.

in the water by the vessel and not the vessel herself which is now under consideration.

But a yacht is not shaped like a rectangular block, nor like a sphere,

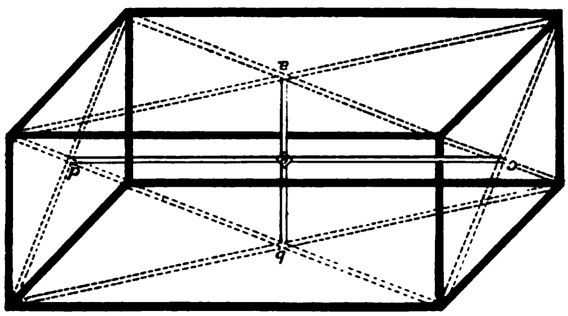


FIG. 4.

and, owing to a yacht's irregular form, the centre of buoyancy is seldom at the mid-length of the hull, nor at its mid-depth; but, inasmuch as both sides of a yacht are, or ought to be, alike, the centre of buoyancy is always

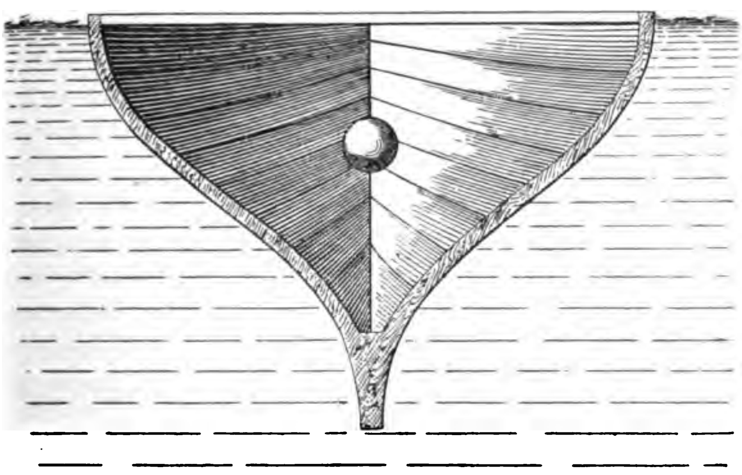


FIG 5.

in the mid fore-and-aft line. Fig. 5 represents a view of the internal under-water space of a yacht, and the ball may be taken as the location of

the centre of the space; in other words, of the centre of buoyancy through which the force represented by the pressure of the surrounding fluid acts.

In yacht designing the principal objects of knowing the fore-and-aft position of the centre of buoyancy is to be able to judge of the relative fulness or fineness of the fore and the after body, and to calculate the position for stowing the ballast or for carrying the lead keel so as to bring the yacht to the designed load line; in the case of steam yachts, it is of use for a similar purpose in determining the positions for carrying ballast, engines, boiler and coal.

Until about the year 1845, it was the practice to place the centre of buoyancy considerably ahead of the centre of length of a vessel; and this was not merely the accidental result of the short full bow and long easy run, as a general conviction prevailed that the centre of buoyancy, by being well forward, tended to secure easy behaviour in a disturbed sea, and prevented a vessel diving: in other words, that it gave her lifting power.

	D Tons.		C. B.
Falcon (brig)........................	434	....	·016 ahead of centre of length.
Vesper (cutter)......................	14	....	·021 ahead of centre of length.
Cygnet (cutter).....................	45	....	·005 abaft of centre of length.

In the second column the longitudinal position of the C.B. is given in terms of the length of the L.W.L.; that is, the distance the C.B. is from the centre of length of the L.W.L. is divided by that length. In the case of the Falcon brig, the centre of buoyancy is 1·7ft. ahead of the centre of length of L.W.L, and the length was 103ft., thus $\frac{1 \cdot 7}{103} = \cdot 016$.

The Pearl cutter, of 137 tons displacement, built at Wivenhoe in the year 1820, for the Marquis of Anglesea, by Sainty, a notorious builder of smugglers, was, however, a remarkable departure from the foregoing, as her centre of buoyancy was as much as 2ft. abaft the centre of length, or expressed as above, ·031, and twenty years later, when Mr. Scott Russell propounded his theory as to what should be the relative lengths of fore and after bodies, the practice became general to place the C.B. further aft; this practice was, however, more the consequence of increasing the length of fore body at the expense of the length of after body, than from any assumed law as to where the location of the centre of buoyancy should be. The Mosquito, built in 1848, was a notable example of the new departure, her centre of buoyancy being ·033 abaft the centre of length of L.W.L.; whilst the America, built in 1851, had hers so far aft as ·049.

At the present day it is common to find the centre of buoyancy still farther aft than in the case in the Mosquito and Pearl, striking

examples being furnished by Buccaneer, Neptune, Queen Mab, Valkyrie III., Isolde, Penitent, &c., all noted racing yachts of great excellence, dating from 1876 to 1896.

	L.W.L. Ft.	C. B.
Buccaneer (Bayly)	34·17	·058 abaft.
Sappho (American)	121·00	·054 "
Neptune (10-tons) (Fife)	39·52	·052 "
Queen Mab (Watson)	59·20	·050 "
America (Steers)	87·25	·049 "
Samoena (Richardson)	80·70	·046 "
Valkyrie III. (Watson)	88·85	·045 "
Isolde (Fife)	59·56	·044 "
Latona (Fife)	93·60	·043 "
Ailsa (Fife)	89·25	·042 "
Windfall (Payne)	33·30	·042 "
Arrow (Chamberlayne)	76·68	·040 "
Vreda (20) (Watson)	45·40	·040 "
Penitent (Payne)	48·00	·040 "
Egeria (Wanhill)	93·80	·039 "
Fiona (Cutter) (Fife)	72·84	·038 "
Mimosa (20) (Clayton)	47·33	·038 "
Vanessa (Hatcher)	47·00	·036 "
Dragon (20) (Fife)	45·36	·035 "
Aline (Nicholson)	100·40	·035 "
Ghost (Clayton)	46·05	·031 "
Vendetta (Payne)	60·4	·030 "
Sleuthhound (Fife)	64·07	·030 "
Constance (now Freda) (Kemp)	82·77	·027 "
Cariad (Payne)	78·4	·026 "
Vol-au-Vent (Ratsey)	79·74	·025 "
Formosa (Ratsey)	82·66	·024 "
Vanduara (Watson)	81·20	·024 "
Freda (20 tons) (Webb)	49·64	·020 "
Miranda (Harvey)	86·35	·026 "
Beluga (Kemp)	66·84	·019 "
Seabelle (Harvey)	90·50	·019 "
Kriemhilda (Ratsey)	79·50	·016 "
Florinda (Nicholson)	85·90	·015 "

Only a general conclusion can be arrived at from an examination of the foregoing particulars, which is, that the centre of buoyancy should be near abaft the centre of length; but it may also be very considerably abaft. As a matter of fact the fore and aft position of the centre of buoyancy does nothing more than indicate the relative fulness of the fore and after bodies; but with the new doctrine, that a long entrance is more useful than a delivery to secure fast reaching power, it became a common conviction that the centre of buoyancy should be far aft. The Arrow can be quoted to justify the conviction. On the other hand, the Florinda, with her centre of buoyancy almost in her centre of length, was noted for speed against any description of vessel of her day (between 1873-84); and further, the centre of buoyancy of Miranda was placed farther aft than Seabelle's by her

designer in the expectation of securing greater reaching power, but the expectation was not realised. Thus it is quite safe to assume that for general performance great licence is permissible in locating the centre of buoyancy in a fore-and-aft direction, and, if situated somewhere near the centre of the length, it is impossible to trace any influence on the speed from slight variations in its position.

The old conviction that a full round deep bow is required to give a vessel lifting power is not now frequently met with; but often a vessel is said to pitch violently in consequence of her heavy after-body, which would, as a matter of course, mean that the centre of buoyancy is relatively far aft. There is no particular reason why a full after-body should of itself cause a vessel to pitch or dive, and these defects may be entirely dependent upon some other cause; but, if a vessel were immoderately full aft, it would entail other bad qualities, such as uneasy twisting motions and yawing in a sea-way, and a general unstableness so far as steering goes. This much however, can be said with some degree of certainty, that the vessels which have been noted for easy fore and aft motion in a sea-way, have had their centre of buoyancy not farther than ·02 abaft the centre of length.

The vertical situation of the centre of buoyancy or its distance below the load water-line, is usually calculated, but knowledge of its position is of little value except in cases where the stability is also calculated.

CHAPTER II.

PROPORTIONS OF YACHTS AND TONNAGE RULES.

Before proceeding to consider the relative proportions of length, breadth, and depth found in yachts, it will be well to trace the influence the operation of the tonnage laws have on these proportions. According to Willet's "Archæologia," vol. xi., a common method of calculating the tonnage of ships prior to 1719 was $\frac{L \times B \times D}{96}$, D being depth in hold. The divisor 96 was probably chosen because it approximately gave the amount of cargo in dead weight a vessel of certain proportions would carry, it being customary to levy dues for vessels carrying coals, &c., by such a method (95 was the divisor used in the United States of America). There would obviously be some difficulty in measuring the depth of hold of a laden vessel; and it seems for this reason, and also to provide against the use of

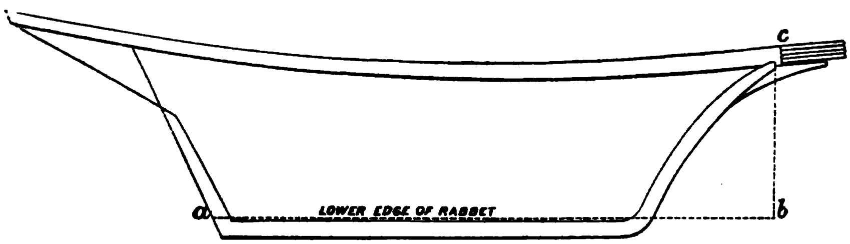


Fig. 6.

false bottoms, that half breadth was substituted for depth to give the cubic measure and the divisor altered to 94, as half breadth was not quite equal to depth in large ships. The length was taken along the keel, and considerable rake of stem and sternpost was usually given a vessel. To allow for this, if the vessel when about to be measured were afloat, $\frac{3}{5}$ of the breadth was deducted from the length to allow for the rake. This rule was confirmed in 1704, with the addition that when the length was taken on deck $2\frac{1}{2}$ inches should be further deducted for every foot of depth at the sternpost for rake, it being assumed that a perpendicular was dropped from the stem and sternpost, and the space measured in from the per-

pendicular to the heel of sternpost and stem in the direction of the straight rabbet line. For small vessels like schooners and cutters the length was still taken along the rabbet of the keel from aft side of sternpost thereat, and an allowance made for rake of stem equal to $\frac{3}{5}$ the breadth. Fig. 6 shows how the length was taken from the back of sternpost along the rabbet line from *a* to *b*, squared down from the stem head at *c*, not including knee or figure head.

It is obvious that by raking the sternpost the length would be shortened, and in yachts as much as 55° rake of sternpost was sometimes met with. This evasion of the tonnage rule led to great discussion thirty years ago, and in 1854 the Royal London Yacht Club took the length on deck instead of along the keel. The Royal Thames Club subsequently adopted this length, and subtracted whole breadth instead of $\frac{3}{5}$ breadth, to satisfy those whose vessels having great rake of sternpost would have had their tonnages raised by a greater number of tons than the vessels with little rake. This rule, having originated in the Thames, was known as "Thames measurement," and was adopted by the Yacht Racing Association in 1876.

$$\frac{(L - B) \times B \times \frac{1}{2}B}{94} = \frac{(L - B) \times \left(\frac{B^2}{2}\right)}{94}$$

When the length on deck from stem to sternpost was substituted for length on keel, some attempts were made to evade it by upright sternposts; but rake aft was found to be a very valuable quality for sailing by the wind and for general handiness, hence rake in a modified degree was always to be found in racing yachts; although in cruisers little or no rake was ever given, and it was the interest of builders to keep the keel as long as possible, as they continued to be paid per ton by the old rule.

About the year 1865, so far as racing yachts are concerned, attempts were made to have the rake and yet evade its consequences by an elbow in the sternpost, as shown by Fig. 7, or by immersing the counter to plumb

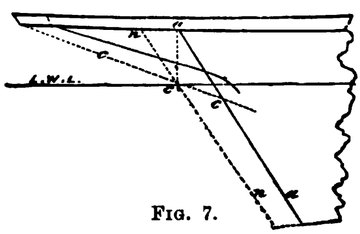


FIG. 7.

with the sternpost on deck. *a a* is a sternpost with counter immersed to *e*; *n n* is a sternpost with elbow from *c* to *a*, the length on deck having thereby been shortened by the piece *n a*. To avoid these evasions the Y.R.A. in 1878 adopted length on L.W.L.

The effect of using beam twice as a factor in the tonnage rule was to place a heavy penalty on it, whilst depth was entirely untaxed. Naturally designers used as little of breadth as they could, and as much of depth. As a measure of tonnage or cubic capacity the rule was entirely fallacious, as it took no cognisance of form. The consequent effect of the rule in the merchant navy was to produce deep box-shaped vessels, slow, and difficult to steer, but carrying a large amount of cargo in proportion to their nominal tonnages. No inducement existed to make yachts box-shaped, but the influence of the rule in producing long, narrow, and deep vessels was most marked.

However, so far as yachts were concerned, the rule for a great many years (up to 1854) fairly enough valued length and breadth, as for any given length a certain proportion of breadth was indispensable to obtain sail-carrying power. It was, even at that period, understood that, other things being equal, length alone governed speed; but length could not be increased without increasing beam in a certain proportion, and so for every foot increase of length a just value was paid for it in tonnage under the rule.

Thus even under a tonnage rule yachts for a time continued to be built broad; when, however, metal keels were introduced it was soon discovered that for any given tonnage length could be increased and breadth decreased to a very considerable extent. This led, in course of time, to vessels being built of five and six beams to length instead of three and three and a half beams; and it became apparent that yachts of even more extreme proportions could be successfully raced against broader yachts of less length. We need not enter into the question as to whether or not a yacht of five or six beams is a better boat than one of four beams; it is sufficient for our purpose to know that for any given tonnage the rule allowed a yacht of six beams to be rated too lightly as compared with the value placed upon one of four or four and a half beams; and the same might be said of five beams as compared with four beams. The effect of this was to produce a continuous exaggeration of one particular costly type of yacht.

So long as beam was absolutely essential for sail-carrying power, too heavy a penalty was not imposed on it; but as the sail-carrying power could be obtained by using an entirely untaxed quantity (depth), length became almost the only element in the hull of a yacht which it was necessary to tax at all. Consequently in 1880 the author proposed to shift the onus of the rating from B^2 to L^2 in the following formula $\frac{L^2 \times B}{1200}$. For a rating by the hull alone, if length only were taxed, there would be a strong inducement to build broad shallow-bodied vessels, with deep and heavily-weighted keels, so as to obtain the advantages of both breadth and depth; hence it was proposed to include breadth in the rule.

After a long controversy a modification of this rule was adopted by the Yacht Racing Association as follows:

$$\frac{(L + B)^2 \times B}{1730}$$

It was soon discovered that this slight modification of the Thames rule had no influence in checking the development of length and the sacrifice of beam; indeed, its chief feature was to precipitate the building of extremely narrow yachts, and so paved the way to an entirely radical change in the rating of yachts for competitive sailing.

The position of affairs in 1886 can be thus summed up: the Thames rule of measurement, and the modifications of it, stood in the way of making any varied experiments in models. The penalty put upon beam, and the absence of any restriction upon depth or ballasting, left, for the purposes of competitive sailing, little opportunity for the naval architect to make extended experiments with form. His ingenuity was almost wholly directed to the question of stability as dependent upon depth and ballasting, and yachts were necessarily of one stereotyped form. These long narrow yachts were capable sea boats, because good behaviour in a disturbed sea is largely dependent upon length and depth; and whilst length was only lightly taxed by the rule, depth was not taxed at all. With this knowledge of the beneficial operation of the rule, it is not surprising that a great resistance was always made to the introduction of any other rule which would check the development of length and depth.

It had several times been suggested that instead of taxing the hull, which has to be driven, the sail, which is the driving power, should be taxed; indeed, sale area formed the basis of comparison for time allowance in the rules of the New York Yacht Club in 1854. It was also advocated by the late Mr. Philip R. Marett for use in this country, and was actually adopted in the season of 1857 by the Royal Yacht Squadron. In the year 1880 the author of this work proposed that length of water line as a prime factor in speed should be used in conjunction with sail as a comparative rating for yachts in competitive sailing, the formula being:

$$\text{Rating} = \frac{L \times S}{6000}$$

In 1882 the Seawanhaka Yacht Club of New York adopted an almost equivalent rule* as follows:

$$\text{Corrected length} = \frac{L + \sqrt{S.}}{2}$$

* An actually equivalent rule would be $\sqrt{L \times S.}$

The New York Yacht Club varied the Seawanhaka rule as follows:

$$\text{Corrected length} = \frac{(L \times 2) + \sqrt{S}}{3}.$$

In the year 1883 the Yacht Racing Association adopted the author's rule as first proposed as an alternative rule, and in 1886 made it the sole rule. Of these rules that adopted by the Y.R.A. admits of the employment of the greatest length for any given rating, as will be seen by the following comparison:

No.	Club Rule.	Length of Water Line. 30ft.	35ft.	40ft.	Corrected Length for Rating.	Rating Y.R.A.
1	Possible sail area New York Y.C. rule $\frac{L\,2 + \sqrt{S}}{3}$	2000 sq. ft.	1225 sq. ft.	625 sq. ft.	35·00 ft.	10
2	Possible sail area under Seawanhaka rule $\frac{L + \sqrt{S}}{2}$	2000 sq. ft.	1578 sq. ft.	1206 sq. ft.	37·36 ft.	10
3	Under equivalent Y.R.A. rule $\sqrt[3]{L \times S}$	2000 sq. ft.	1714 sq. ft.	1500 sq. ft.	39·15 ft.	10
4	Possible sail area for 10-rater under Y.R.A. rule $\frac{L \times S}{6000}$	2000 sq. ft.	1714 sq. ft.	1500 sq. ft.	—	10

It will be seen that the operation of this rule may affect the proportions of yachts in quite a diversified manner; and one general effect was to make yachts broader for any given length than they were prior to 1886. Speaking generally of the influence of the rule it can be said that if for any given rating a yacht was short she could have a large sail spread and required great beam or depth of ballasting or weight, or a combination of all three, to enable her to carry the sail.

If the yacht is long in proportion to her sail spread her displacement is required to be reduced to that minimum which will just enable her to carry the sail due to her length and rating.

However, in this country, at any rate, short yachts with large sail spreads did not answer in average weather, and the final outcome of the rule was to produce yachts of small sail spreads in proportion to their lengths, the comparison being made by

$$\frac{\sqrt{\text{Sail Spread.}}}{\text{L.W.L.}}$$

As a tax on sail is practically a tax on displacement, the yachts became very shallow in body, with excessive draught of water, and finally the fin-bulb keel (of which examples will be found among the plates) was adopted.

This type, on account of the small head room in yachts of 64ft. L.W.L. and under, was as much objected to as the narrow, deep-

bodied craft built under the old tonnage rule, and in 1895 the Yacht Racing Association, assisted by the leading yacht designers and Mr. R. E. Froude, F.R.S., adopted a hull girth measurement (based on Lloyd's rule for scantlings), combined with length of water line, beam, and sail spread as follows :

The rating of every yacht entered to sail in a race shall be ascertained by adding together length (L), beam (B), 0·75 of girth (G), and 0·5 of the square root of the sail area (SA), and dividing the sum by 2 according to the following formula :—

$$\frac{L + B + 0{\cdot}75\,G + 0{\cdot}5\sqrt{SA}}{2} = \text{Rating.}$$

In all ratings, figures in the second place of decimals below 0·05 shall be disregarded, and those of 0·05 and upwards shall count as 0·1.

The length shall be taken between the outer edges of the Official marks of the Y.R.A., as placed by the owner at the bow and stern of the yacht, this length to represent the extreme length for immersion, provided always that if any part of the stem or stern-post or other part of the vessel below the marks for length project beyond the length taken as mentioned, such projection or projections shall, for the purposes of the rule, be added to the length taken as stated; and pieces of any form cut out of the stem, stern-post, or fair-line of the ridge of the counter, with the intention of shortening the length shall not be allowed for in measurement of length if, at or immediately below the marks for the length, nor above if within 6 inches of the water level.

The breadth shall be taken from outside to outside of the planking, in the broadest part of the yacht, and no allowance shall be made for wales, doubling planks, or mouldings of any kind.

The girth shall be taken from L.W.L. to L.W.L. under the keel at a station 0·6 of the distance between the outer edges of the length marks from the fore-end. The girth shall be measured along the actual outline of the vertical cross section at that station at right angles to the L.W.L., *see* Figs. 8 and 9, *a a′*, and Figs. 10 and 11. If the draught forward of that station,

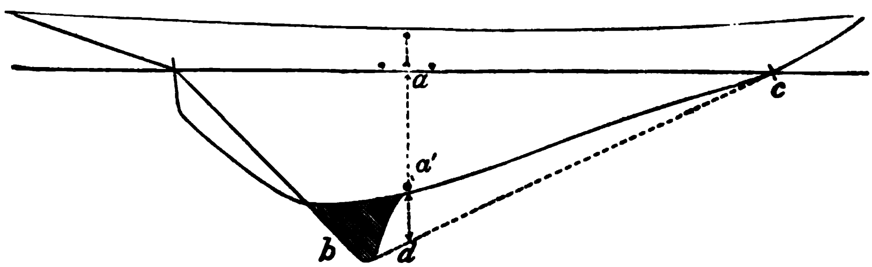


FIG. 8.

_e e_, Fig. 9 (not including the girth of a bulb, if any) exceeds the draught at that station, *a a′*, twice such excess to be added to G. In taking these measurements all hollows in the fore and aft under-water profile of the vessel to be treated as filled up straight. Should a piece be added, as at *b*, then a line must be drawn from *b* to *c*, and the girth be measured to *d*. Fig. 8.)

In the case of a centre-board, 1·5 times the extreme drop of the board below the keel to be added to the girth as taken at *a a′*; and if the board is dropped below the keel at *e e*, the excess at *e e* shall, nevertheless, be added to girth in accordance with the rule. Bulb or ballasted centre-boards to be measured as fixed keels.

Owners shall mark the length for rating of their Yachts on both sides at the bow and stern in such manner as the Council may direct, with the official marks supplied by the Y.R.A., which marks shall at all times represent the extreme length for immersion when the yacht is lying in smooth water in her usual racing trim, including racing crew on board at and about the mid overall length.

Owners shall mark the points for measuring the girth, as follows : by fixing three metal discs of suitable size on each side of the yacht, not less than 2 inches or more than 6 inches

above the load water-line level (and parallel thereto), and not less than 3ft. or more than 6ft. from end to end, and so that the centre mark, *a*, of the three, coincides with the distance 0·6 from the fore edges of the bow marks (*see a* and *c*, Fig. 8); and the owner shall also place a disc coinciding with this centre mark (perpendicular to the load water-line level), under the rail

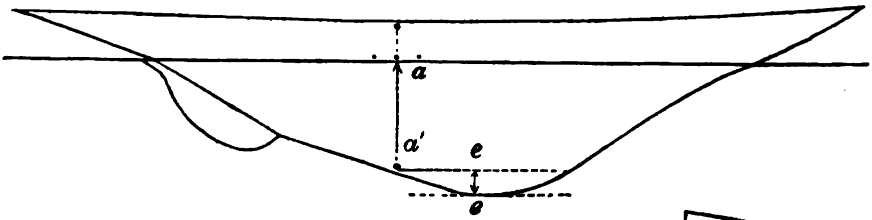


Fig. 9.

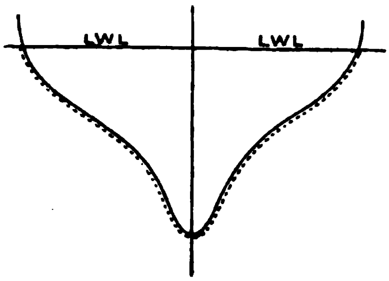


Fig. 10.

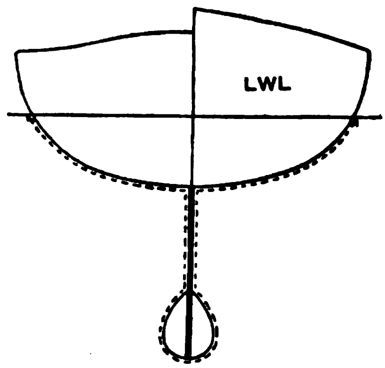


Fig. 11.

or covering board and another on the side of the keel perpendicular to the load water-line level (*see a'* Figs. 8 and 9). The distances between the load water-line level and the horizontal marks to be measured when the yacht is afloat in smooth water with crew on board according to the rule, and deducted from the girth as obtained from centre mark to centre mark.

Example of working:

L.W.L. = 45·6ft.	Girth = 23·4.
Beam = 13·0.	Sail Area = 2600.

The girth multiplied by ·75 is 17·55.
The square root of the sail area 2600 is 51 which multiplied by ·5 equals 25·5.
Then the sum will be:

L..............................	45·60
B..............................	13·00
·75 G..........................	17·55
$\cdot 5\sqrt{SA}$	25·50
	2)101·65
	50·87 = 50·9 linear rating.

Of course the object aimed at in this rule is to induce fuller mid-sections, as by filling them out the girth would be shortened; and, as only half the square root of the sail area is used in the formula, this is expected to induce larger sail spread, which, in turn, is expected to encourage a larger under water body to carry the sail, and the greater driving power may prevent a falling off in speed. At present (1896) the general expectation regarding the influence of the rule appears likely to be realised, and forms prevalent under the first operation of

the $\frac{L \times S}{6000}$ rule of 1886 are coming into fashion again, but with much larger sail spreads; however, it is too early to speak about the new formula with precision, especially as many of the shallow fin-bulb type are successful.

With regard to cruising yachts—or yachts built entirely with a view of getting the most possible comfort whilst afloat—it cannot be contended that the old tonnage rule necessarily operated against beam. In short, it was the builder's interest to make the vessel as broad as possible, as he was generally paid by the ton, and under this rule beam increases tonnage nearly eight times as fast as length, if the proportion of length to beam is as five to one. But as a matter of fact the person for whom the yacht was to be built knew this condition of the rule as well as the builder; and as 8ft. of length is of more value for internal accommodation than 1ft. of breadth, it is not surprising for this reason alone that owners were willing to put up with as little beam as possible. For any given tonnage the longer and deeper vessels are more costly to build, and the price per ton was necessarily increased by the builders, a fact which owners were quite aware of, but it was generally attributed solely to the increased price of material and labour.

However, for cruising, a yacht has generally been given a greater proportion of beam to length than a racing yacht; but the proportion which beam should bear to length is subject to a great variety of opinions. The true reasons for giving a cruising yacht more beam than a racing yacht were these: In the first place initial stability, or resistance to be heeled from an upright position at the commencement of inclination, can be attained by beam at a less cost than by depth heavily weighted with lead. It is very desirable for comfort that a cruising yacht should be kept as upright as possible; for cruising, nothing can be more uncomfortable and inconvenient than for a yacht, even in moderate breezes, to be always over on her side: it will be impossible to walk about her deck, and the discomfort and disorder below are unbearable.

The results of practice indicate that for a cruising yacht the length should be from 4 to 4·75 times the beam, or, in other words, beam should equal length on load line multiplied by factors ranging from ·25 down to ·211.

With regard to the height of freeboard* a great difference of opinion would appear to have prevailed, inasmuch as yachts of 150 tons exist with only 2ft. 9in. of side above water at the lowest point of the deck, and yachts of 40 tons are frequently given as much as three feet. One rule for determining freeboard is based on the assumption that a yacht should have so much side that when she is heeled none of her deck is immersed until a certain angle of heel is reached. This angle was generally put at 30° for small yachts of 5 tons, and diminished to 24° for yachts of

* Freeboard here means the least height of side from the water surface to the deck.

larger tonnage. This rule is a trustworthy one so long as the yachts are of from four times to four and a half times their beam in length; but when the length much exceeds these proportions it is inapplicable. For instance, the Jullanar is nearly six times her beam in length, or 16·88ft. to 100ft., and if her freeboard had been apportioned by the rule that her deck was to reach the water at 24° inclination, she would only have been given 3ft. 2in. of side at the lowest point. This would have been a sufficient proportion for a yacht, say 72ft. long and 16ft. broad, equal to 77 tons; but the Jullanar is 27ft. longer, and has the propelling or heeling force in the way of sails usually given a yacht of nearly double 77 tons, whilst her stiffness is not in that proportion; she therefore requires increased side to assist her stability so that she may carry her canvas effectively, and at the same time keep her as dry in a seaway as the shorter and slower yacht. (*See* chapter on "Stability.") It is thus quite plain that the freeboard of a yacht should not be decided upon by beam alone, irrespective of length; neither would it be satisfactory to decide upon freeboard by length without reference to beam, and a rule is required which will include the proportion of beam to length, and maintain approximately an equal volume above water for any given length. It is found that the freeboard of well-proportioned yachts of four and a half beams to length varies as the 1·8* roots of their breadths multiplied by a factor varying with the proportion of breadth to length on the load water line. Freeboard $= \sqrt[1\cdot8]{\text{Beam}} \times$ Factor.

The following tables will supply factors for yachts of all proportions and sizes:

For Yachts exceeding 25ft. Water Line.

No. of Beams to length.	Factor.	No. of Beams to length.	Factor.	No. of Beams to length	Factor.
3	·50	4¾	·75	6½	·96
3¼	·54	5	·78	6¾	·99
3½	·58	5¼	·81	7	1·02
3¾	·62	5½	·84	7¼	1·04
4	·66	5¾	·87	7½	1·06
4¼	·69	6	·90	7¾	1·08
4½	·72	6¼	·93	8	1·10

For Yachts or Boats not exceeding 25ft. Water Line.

No. of Beams to length.	Factor.
2½	·60
2¾	·64
3	·68
3¼	·71
3½	·74
3¾	·77
4	·80
4¼	·83
4½	·86
4¾	·89
5	·92

For Steam Yachts.

No. of Beams to length.	Factor.
5	·65
5¼	·67
5½	·69
5¾	·71
6	·73
6¼	·75
6½	·77
6¾	·80
7	·83
7¼	·87
7½	·91
7¾	·95
8	1·00

* A table of the 1·8 roots of numbers will be found in the Appendix.

The rule applied to a narrow yacht of $5\frac{1}{4}$ beams as an example, like the yawl Lenore, 89ft. on the water line, and 17·2 beam, would give her $\sqrt[1\cdot8]{17} \times \cdot81 = 4$ft. The exact freeboard of Lenore at the lowest point is 4·2ft. For the 20-rater Vreda we have $\sqrt[1\cdot8]{10\cdot1} \times \cdot75 = 2\cdot7$ft., which is her exact freeboard. The 13ft. Itchen punt Vril has a freeboard of 1·4ft., and by the formula it would be $\sqrt[1\cdot8]{5\cdot2} \times \cdot6 = 1\cdot5$ft.; in fact, the rule, if applied to yachts of any extreme proportions, up to lengths of 200ft. water line, will give a suitable height of freeboard for them.

So far as steam yachts are concerned, the practice now is to give them considerably more freeboard than was formerly the case, and they are in consequence much drier on the deck and safer in bad weather. They do not, however, require quite as much freeboard as sailing yachts of equal length, and as a rule they have higher bulwarks.

The foregoing formula and table will give a suitable light load water-line freeboard for a steam yacht; that is what her freeboard should be when she has burnt all her coal.

In designing merchant ships it is usual to consider freeboard in relation to depth as well as length and breadth, but as a yacht's free-board may be regarded as an unalterable quantity after she is once ballasted (excepting in the case of steam yachts), depth under water need not enter into the calculation, as the quantity of freeboard given by the rule is sufficient for any depth a yacht is likely to be given be she steam or sail, and is at the same time appropriate for shallow yachts; indeed, the freeboard might be reduced for very deep yachts without unfavourably influencing their performances, in order to keep the weight of deck, &c., as low as possible, as well as the centre of effort of the sails, but in shallow yachts freeboard becomes an element in insuring safety, as it largely assists in lengthening out the range of stability, as will hereafter be shown.

The draught of water of cruising yachts is limited to the requirements of coasting, and whilst a yacht, say of 60ft. on the load line, can have a draught of 11ft., a yacht of double length cannot conveniently have more than 14ft.

The draught of water of yachts has steadily increased since 1870, but at the same time the area of the middle vertical fore and aft plane has decreased. Thus in 1876 we find a 20-tonner like Challenge 47·5ft. on the water-line, with an extreme draught of 8·5ft., and area of fore and aft vertical plane of 360 sq. ft., giving a mean draught $\left(\frac{\text{Area L.W.L.}}{\text{Length.}}\right)$ of 7·2ft. In 1890 the Ghost, 20-rater, 46·5ft. on the water line, had an extreme draught of 9·8ft. and a fore and aft vertical plane area of 300 sq. ft., giving a mean

draught of 6·45ft. only. In 1894 the draught for the 20-raters ranged from 10ft. to 11ft.

In larger yachts, like Kriemhilda, 80ft. on the water line, the extreme draught was 12·1ft., and the area of vertical fore and aft plane 847 sq. ft., giving a mean draught of 10·6ft. In the case of an 85ft. racing yacht, built in 1887, the extreme draught is 13ft., and the area of fore and aft vertical plane about 850 sq. ft., giving a mean draught of 10ft.

It might have been inferred that the draught would have varied in a ratio with the square root of the area of vertical fore and aft plane, but, owing to the fact that small yachts have a greater proportional draught, as already described, this is not the case. According to present practice, however, the draught varies pretty regularly as the Length$^{\frac{2}{3}}$ × ·75 for cruisers, and as Length$^{\frac{2}{3}}$ for racers, as set forth in the table.

Length on Water Line in Feet.	*Length$^{\frac{2}{3}}$	Factor for Cruisers.	Extreme Draught of Water in Feet.	Factor for Racing Yachts.	Extreme Draught of Water in Feet.
20	7·4	·7	5·2	·8	5·9
25	8·5	·7	6·0	·8	6·8
30	9·6	·7	6·7	·8	7·7
35	10·7	·7	7·5	·8	8·5
40	11·7	·7	8·2	·8	9·3
45	12·7	·7	8·9	·8	10·1
50	13·6	·7	9·5	·8	10·9
55	14·5	·7	10·3	·8	11·6
60	15·3	·7	10·7	·8	12·2
65	16·1	·7	11·3	·8	12·9
70	17·0	·7	11·9	·8	13·6
75	17·8	·7	12·5	·8	14·2
80	18·6	·7	13·0	·8	14·9
85	19·3	·7	13·5	·8	15·4
90	20·1	·7	14·1	·8	16·0
95	20·8	·69	14·3	·8	16·6
100	21·5	·68	14·6	·8	17·0

It must be clearly understood that this is not an attempt to set up an arbitrary value for draught in yachts of varied proportions and sizes. But the formula being the outcome of the examination of many yachts of different sizes, gives approximately the appropriate draught of water a vessel should have according to existing fashion, unless the draught is controlled by some arbitary considerations.

Neither has the rule any scientific pretensions, as it is not based upon any mathematical investigation, but simply gives results agreeable to present practice, which practice is the latest step in the progressive stages

* The $\frac{2}{3}$ power is the cube root of the square of the number, or the cube root squared For instance, the cube root of 27 is 3, and the square of 3 is 9, hence 9 is the $\frac{2}{3}$ power of 27.

the proportions of yachts have passed through; consequently, in the course of time, such a rule may be found to be at variance with practice.

TABLE SHOWING HOW THE NEW RATING AFFECTS EXISTING YACHTS.

Name.	Y.R.A. Rating.	L + B + ·75 G + ·5 $\sqrt{SA}$.				Total.	Extreme Draught.	
							FT.	IN.
Satanita	164	97·80	24·5	29·61	51·18	203·09	16	3
Calluna	140	84·00	24·4	30·0	50·0	188·4	14	5
Iverna	118	84·00	19·0	24·0	45·98	172·98	13	0
Irex	98	84·00	15·0	23·0	41·77	163·77	13	0
Blue Rock	62	64·50	17·5	24·2	38·00	144·20	12	5
Silver Star	59	66·00	11·50	20·25	36·04	133·79	11	7
Annasona	42	64·20	11·9	18·40	31·10	125·6	10	7
Thalia	40	59·10	14·0	21·10	31·8	126·0	12	7
Lais	40	60·00	17·0	22·0	31·6	130·6	11	6
Deerhound	40	58·80	13·5	19·9	31·98	124·18	11	6
Creole	40	59·50	13·2	20·4	31·75	124·85	12	1
Queen Mab	40	59·70	16·3	19·30	31·70	127·00	10	9
Varuna	40	59·00	14·6	21·60	31·88	127·08	13	3
Carina	40	60·00	15·75	23·00	31·60	130·35	12	9
Corsair	40	58·70	14·50	21·40	32·00	126·60	13	3
Vendetta	40	60·40	17·00	23·55	31·50	132·45	11	9
Inyoni	20	43·60	13·10	20·00	26·2	102·90	9	9
Stephanie	20	46·53	12·25	21·00	25·33	105·11	11	2
Chiquita	20	45·50	10·91	16·43	25·60	98·44	9	9
Deirdre	20	47·00	12·80	19·50	25·26	104·56	11	3
Asphodel	20	45·50	12·35	18·10	25·67	101·62	9	10½
Zinita	20	46·10	13·10	16·80	25·50	101·50	9	1
Luna	20	45·60	13·00	18·70	25·60	102·90	9	8
Thelma *	20	45·70	13·50	21·40	25·60	106·20	13	0
Lilith	10	36·00	10·60	15·00	20·40	82·00	7	6
Kite	10	36·10	8·60	13·60	20·30	78·60	8	7
Doris	10	33·50	6·20	11·70	21·20	72·60	7	0
Phantom	10	35·00	7·92	13·70	20·70	77·32	8	4
Dora *	10	35·40	10·20	15·20	20·60	81·40	10	6
Decima	10	36·10	9·91	15·18	20·40	81·59	8	6
Rosetta	10	34·90	10·50	18·60	20·70	84·70	8	8½
Dacia	5	35·00	8·20	14·9	14·60	72·7	8	3½
Flat Fish	5	32·00	10·00	17·31	15·33	74·64	7	6
Red Lancer	5	30·90	9·40	11·20	15·50	67·00	6	5
Fleur-de-Lis	5	31·70	9·00	14·60	15·30	70·60	6	6½
Delanagh	5	31·50	8·60	16·20	15·40	71·70	7	0
Natica	5	32·50	8·62	13·70	15·2	70·02	7	9
Windfall	5	32·80	7·80	13·43	15·27	69·30	8	0
Papoose	2·5	26·70	7·25	12·09	11·71	57·75	5	7
Faugh-a-Ballagh	2·5	27·40	6·92	11·12	11·64	57·08	6	7½
The Babe	2·5	26·80	6·75	10·50	11·65	55·70	5	11
Kismet	2·5	24·20	7·12	14·55	12·05	57·92	6	9
Humming Bird	2·5	25·70	7·41	10·68	11·90	55·69	5	10½
Mystery	2·5	26·10	7·40	13·60	11·90	59·0	6	6½
Bulbul	2·5	25·40	6·50	9·90	12·1	53·9	6	2½
Dolphin	2·5	26·10	7·50	11·10	11·82	56·52	5	9
Ferida	2·5	27·50	7·00	12·90	11·70	59·10	6	11
Gareth	2·6	28·90	6·90	13·30	11·60	60·70	6	8½
Wolfhound	1	19·14	7·30	12·44	8·85	47·73	5	7½
Fay	1	20·50	7·16	12·00	8·58	48·24	5	0
Sacharissa	1	19·50	5·66	10·43	8·44	44·03	4	3
Estrella	1	19·40	5·29	8·81	8·80	42·30	4	6
Red Rover	1	20·00	5·66	9·00	8·65	43·31	5	2
Tip Cat	1	20·00	6·00	10·62	7·79	44·41	4	3

* Centre-board. Thelma's draught without centre-board is 8ft. 7in.

CHAPTER III.

PROPORTIONS, FORM, WEIGHT, AND BALLASTING OF YACHTS AND STABILITY.

The weight of water which a yacht or vessel of any kind displaces is equal to her own weight, and consequently the pressure of the water on the immersed vessel is equal to her own weight. This pressure is diffused all over the immersed part of the hull, and the pressure of any individual particle of water on the hull is in a direction at right angles to the point of contact. The concentrated pressure, or the resultant of the pressure, on the immersed portion of the hull, acts vertically in a line produced through the centre of buoyancy; and, as before said, this pressure is equal to the weight of the ship. Thus there are two equal forces acting in opposition to each other, and balancing each other—the weight of the

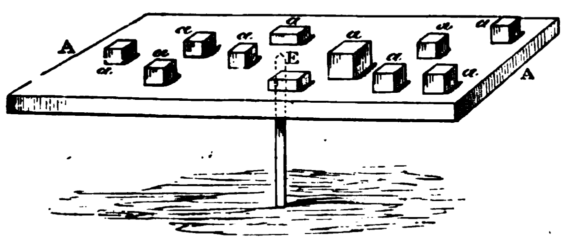


Fig. 12.

displaced water pushing upwards through the centre of buoyancy, and the weight of the ship pushing downwards through its centre of gravity; and these two forces never act in any other than a vertical direction.

The centre of gravity of a yacht or ship is a determinable point, where the action of all her weights is concentrated; therefore it is sometimes called, in relation to ships, "the centre of gravity of the whole mass." The "whole mass" includes the hull, ballast, spars, sails, fittings, crew, stores, and everything which the ship or yacht contains that is of any weight at all. If on a plank A (Fig. 12) a number of weights, *a a a a*, &c., be placed at any irregular or equal intervals, and the plank be made to balance on a pointed stake at E, then E will be the common centre of

gravity of the plank and all the weights placed upon it. Thus the exact position of the centre of gravity of a ship depends upon the disposition of her weights—no matter whether these weights be timbers, keel, plank, ballast, spars, rigging, sails, crew, stores, or anything else that is of weight—and it follows in a ship that, if the weights are placed further forward, the centre of gravity will be shifted forward, and the contrary if the weights be shifted aft. In a like manner, if the weight of the masts, sails, or gear be increased, the centre of gravity, with regard to its vertical position, will be brought higher; on the other hand, if the ballast be increased in weight, or if it be stowed deeper in the hull, or if the lead keel be lowered, the centre of gravity, with regard to its vertical position, will be carried lower.

Thus we have two distinct, but balanced, forces—the weight of the water the ship displaces acting upwards through the centre of buoyancy, *k*

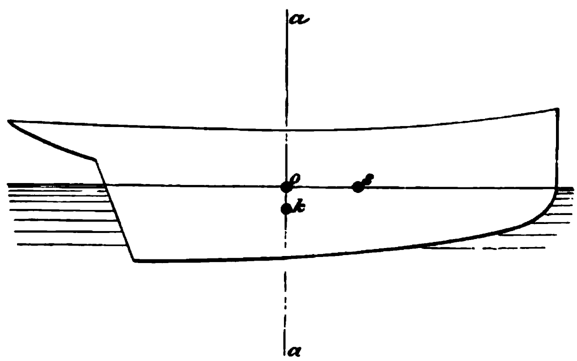


FIG. 13.

(Fig. 13), and the weight of the ship acting downwards through *o*, its centre of gravity.

A necessary condition for the ship to be in equilibrium is that the resultant of the two forces, represented by the weight of the ship acting through *o*, and the weight of the water she displaces, acting through *k*, should have effect in the same vertical line. (See *a a*, Fig. 13.)

If the direction of action of either be shifted, a struggle will instantly commence to regain a position where they will balance each other again, or act in the same vertical line. For instance, let a portion of a yacht's ballast or other weight be shifted forward until her centre of gravity is shifted from *o* to *s*, Fig. 13, then the yacht will sink down by the head until the two forces are directly over each other again, as *s* k^1, in the vertical line *b b*, Fig. 14, and where two lines *a a* and *b b* cut each other at M is termed the metacentre, or for fore and aft motion the longitudinal metacentre.

Now if the *centre of buoyancy* had been carried to k^1 by the vessel being hove down by the head otherwise than by having a portion of her ballast or weight moved forward, such, for instance, as by a pressure of wind on her sails, she would instantly regain the position depicted in Fig. 13 upon the removal of the force or pressure which had hove her down. This effect that brings a vessel back to her original condition of equilibrium is called her righting power, or statical stability; and for the motion we have described would be termed her longitudinal statical stability. When a vessel is placed among waves, the centre of buoyancy is continuously carried forward or aft, as she is differently water-borne by the passing waves. A constant struggle is thereby maintained between the centre of gravity of the vessel and her centre of buoyancy to keep in the same vertical line *a a*, and an uneasy violent motion is acquired, the force of which, to some extent, is dependent upon the urgency of the righting power.

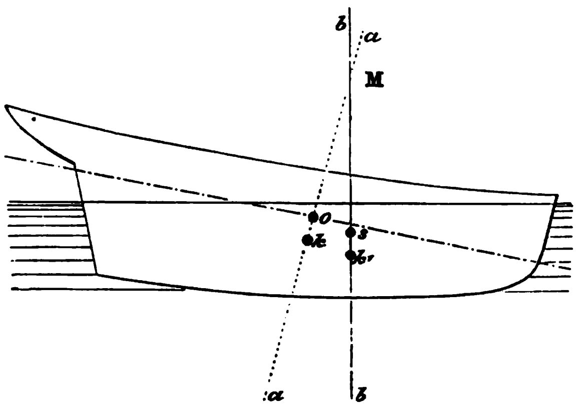


FIG. 14.

A vessel with her weights or ballast stowed low will have this righting power in a greater degree than one with her weights stowed high; and she will be relatively quick in "recovering herself." So also will be a vessel that is very full on the load water-line, and very much cut away underneath; whereas a vessel with what is known as a long body will be comparatively easy in her motions during similar wave disturbance. The pitching and 'scending motions of vessels form a very complex problem, and are by no means wholly dependent on the conditions just adverted to. For instance, the *momemtum* acquired during pitching, whilst the bow is left unsupported by the water, or 'scending, whilst the stern is without support, may be much increased by the distribution of the weights or ballast in a fore-and-aft direction, as the radius of gyration would be thereby lengthened; at the same time a vessel's quickness in recovering herself after pitching would be decreased and she might ship much water;

but the influence on these motions, which any particular condition of a vessel may have, can only be determined in a general kind of way, and are not amenable to any precise calculation. This subject will be further dealt with in a subsequent chapter.

Hitherto reference has only been made to the longitudinal or fore-and-aft motions of a vessel, but as a yacht's righting power or stability is generally spoken of in connection with her heeling or rolling, it will be best to illustrate it in connection with these transverse motions.

Fig. 15 is a representation of a transverse section of a vessel supposed

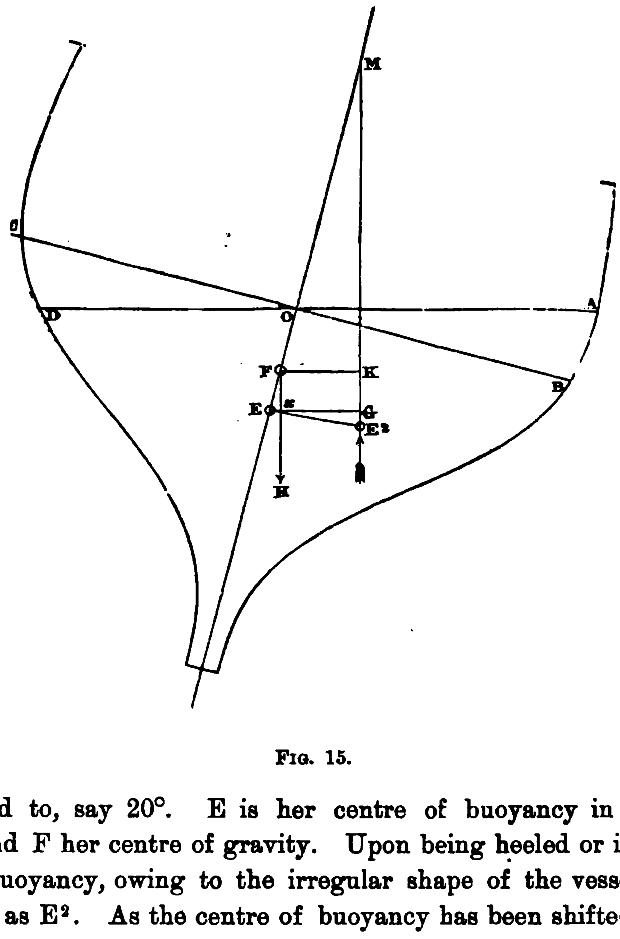


Fig. 15.

to be heeled to, say 20°. E is her centre of buoyancy in an upright position, and F her centre of gravity. Upon being heeled or inclined, the centre of buoyancy, owing to the irregular shape of the vessel, shifts to some point, as E^2. As the centre of buoyancy has been shifted to E^2, the resultant of the water pressure no longer acts through E, but through E^2; and it must be remembered that this resultant always acts vertically, or at right angles to the water-level. The resultant of the force represented by the weight of the ship continues to act vertically downwards through her centre of gravity F, that being the point it would act through if the

vessel had not been heeled; and it is assumed that no part of the weight of the ship has been shifted, so as to cause her centre of gravity to shift.

The wedge-shaped piece of the hull A B O is called the wedge of *immersion*; and the wedge-shape piece O C D the wedge of *emersion*. By naval architects they are usually referred to as the "in" and the "out" wedges. Owing to the volume of displacement on the immersed side of the middle vertical line gradually increasing in bulk, whilst the emersed side is decreasing as the vessel becomes inclined, the centre of buoyancy shifts over to leeward, and so the couple x G (Fig. 15) is formed. The first resistance to inclination is usually referred to as initial stability, a quality which broad and shallow vessels have in excess.

In saying that the volume of displacement on the immersed side increases in bulk as the vessel becomes inclined, whilst the part on the emersed side decreases, it must not be supposed that the displacement is increased in proportion to the excess on the immersed side. *The displacement always remains exactly the same as the weight of the vessel;* but if the volume of immersion be in excess of the volume taken out, then the vessel shifts or rises bodily in the water, to an extent which is dependent upon the area of the new load water-plane and the excess in the volume of immersion.* ?

It will be assumed that a vessel has been heeled to an angle of say 15° by some force such as that of the wind on the sails. Upon reference to Fig. 16 (on the page over leaf) let L.W.L. be the load water-line of the upright position; then upon the heeling of the vessel the wedge-shaped piece A B O is put into the water, and the piece C D O taken out. If the cross sections of the vessel were circular in form, the wedges A B O and C D O would be exactly equal; but as the cross sections are of an irregular form the wedge of immersion, represented by A B O, is apparently in excess of the wedge of emersion C D O. Now it is obvious that the displacement of a ship (always equal to her weight) cannot be added to by simple inclination, and therefore in reality the wedges of immersion and emersion are equal. To make them equal for the purpose of calculation of stability at small angles of heel a line shown by the ticked line $a\ a$

* A yacht when sailing through the water at great speed may apparently, and does actually, sink below the general water level owing to the large hollow wave amidships; but the crests of the bow and stern waves to a large extent balance the hollow amidships, and were it not for this, there might be a serious deficiency in stability. Beyond this the motion of the vessel through the water would influence stability; and, again, it would also influence heeling force, or wind current applied to the sails. This matter is discussed by Mr. W. H. White in his "Manual of Naval Architecture;" and he suggested that experiments should be tried with yachts as to their stability when moving through water, and the heeling effect of the wind when applied at various angles. The same idea occurred to the author, but the difficulties in the way appear insurmountable. ?

(Fig. 16) is drawn, and then the volume of the immersed and emersed wedges represented by A B *b* and C D *b* are calculated. It is quite possible that two or three lines will have to be tried before the wedges are found equal, but the distance O *b*, the point of intersection of the load water-line of the upright position with the "trial" line of the inclined positions, can be assumed to be O *b* = 0·25ft. × degrees of angle of heel.

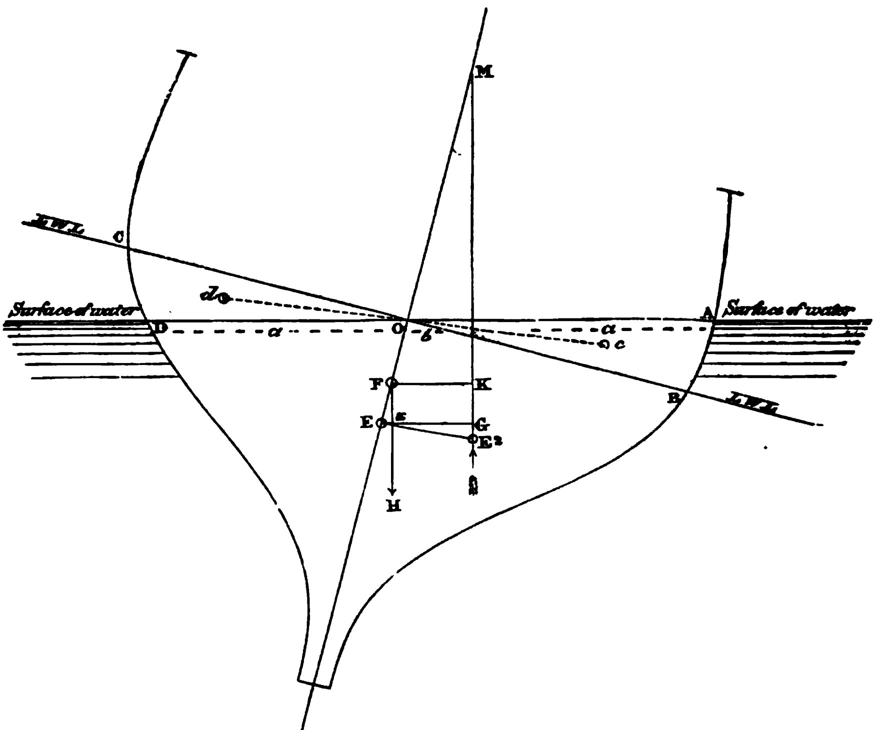


FIG. 16.

When heeled, as previously explained, the centre of buoyancy, E, shifts to some point, E^2, and this point can be determined thus: Let *c* be the centre of gravity of the wedge of immersion, and *d* the centre of gravity of the wedge of emersion, both wedges being equal; then the distance *c d* multiplied by the volume of the wedge, and divided by the whole displacement of the vessel, will give the distance the centre of buoyancy has shifted, E^2.

$$E^2 = \frac{c\,d \times \text{volume of wedge.}}{\text{Displacement.}}$$

From the original centre of buoyancy the distance found, as explained and represented by E E^2 (Fig. 16), must be drawn parallel to *d c*; then the upward pressure of the water will act through E^2 in the direction of E^2 M, at right angles to the line A D, or surface of water. The force repre-

sented by the weight of the ship will act downwards, through the centre of gravity, F, at right angles to A D in the direction of the line and arrow F H. From the original centre of buoyancy produce E G at right angles to E^2 M, and parallel to A D. This line will intersect F H at x, and the distance represented by x G = F K will be the length of the righting lever, which, multiplied by the displacement in tons, will be equal to the righting moment in foot tons.

If the centre of gravity were still at E (the original centre of buoyancy), it is plain that the length of the lever would be E G instead of x G; therefore, inasmuch as the centre of gravity lies above E, the righting moment is less E G by the quantity represented by E x; or the righting moment is E G × displacement in tons − E x × displacement in tons = x G × displacement in tons.

The intersection at M is the metacentre,* and is the point where the middle line of the vessel when in an upright position is cut by the line produced through the new centre of buoyancy, E^2, of the inclined position. In a vessel of cylindrical form the metacentre M is a fixed point; in other cases it shifts according to the form and inclination of the vessel. The proper term, therefore, is shifting metacentre for ordinarily shaped vessels; however, in most vessels the amount of its shifting is immaterially small for angles of inclination up to 10°; for such inclinations the metacentre is assumed to be a fixed point, and a common expression for the righting moment at any small angle of heel is the height the metacentre is above the centre of gravity (or the distance F M) multiplied by the displacement in tons, multiplied by the sine of the angle of heel. Righting moment in foot tons = F M × displacement × sine θ. The height of the shifting metacentre above the centre of buoyancy E^2 (see Fig. 12).

$$E^2\,M = \frac{d\,c \times \text{volume of wedge.}}{\text{Displacement} \times \text{sine } \theta \text{ heel.}}$$

The distance the centre of buoyancy E has shifted from E_2 is thus found

$$E\,E^2 = \frac{c\,d \times \text{volume wedge.}}{\text{Displacement.}}$$

The mean volume of the two wedges is used, that is to say, the volume of one wedge is, say, 713 cubic feet, and that of the other 719 cubic feet, the mean of these two would be 721 cubic feet, which would be the quantity used in the calculation.

* The height of M above the centre of buoyancy is determined by cubing the ordinates of the L.W.L. in the half-breadth plan and dividing the sum by the displacement in cubic feet; hence adding to the displacement without increasing the breadth at the L.W.L. will decrease the metacentric height. This will appear in the calculations farther on.

The distance E E^3 is set off from E *parallel* to the line $c \cdot d$. However, in order to find the length of the righting lever x G (Figs. 11 and 12) it is not essential to know the position of E^3, but only the direction of the line through E^2 M at any angle of keel, and this will be demonstrated in the examples of the method used in calculating the centres of the wedges given further on.

The righting moment or power is computed by multiplying the weight of the ship, or displacement in tons, by the length of the righting lever x G (Figs. 11 and 12). That is, if the weight of the ship or her displacement be 40 tons, and the length of the righting lever at 20° inclination be 2ft., then her righting power or moment of stability at that inclination will be $40 \times 2 = 80$ foot-tons. If the righting moment of a yacht at 20° inclination be equal to 80 foot-tons, as described, then it will require a steady moment equal to 80 foot-tons upon her canvas to maintain her at that inclination.

If the height the point M (termed the metacentre) above the centre of gravity, G, be known for a small angle of heel, the length of the righting lever x G can be calculated by the following equation: M G $\times$ sine of angle of heel. *See* example farther on.

It is quite a common thing to hear a person say that this, that, or the other vessel has "great artificial, but very little natural or structural stability," as if there were various kinds of stability. This confused way

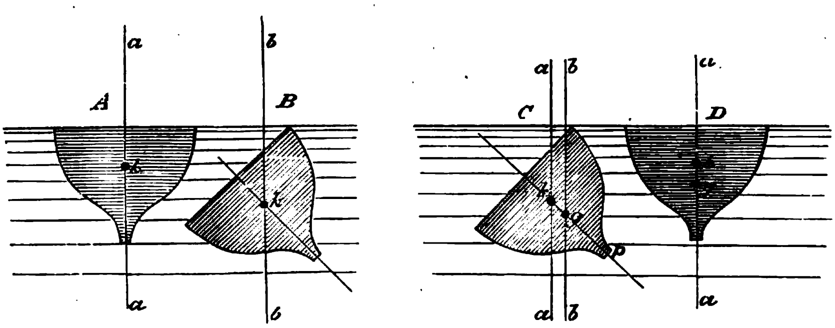


FIG. 17.

of regarding stability is very likely to prevent a clear understanding of the conditions on which stability depends, and it must be understood that there are no such things as "artificial stability" or "natural stability" or "structural stability" or "stability of form" as distinct qualities.

It may be assumed that a homogeneous substance is placed in a fluid, or that a portion of a fluid is turned into a solid, maintaining its inherent bulk, weight, and uniform specific gravity; then such a substance or solid would rest in whatever position it were placed. Let A (Fig. 17)

be such a substance; then its centre of buoyancy and centre of gravity must necessarily be at the same point, *k*; and, as the resultant of these two forces acts in the vertical line *a a*, the body will be in equilibrium if placed in the position—which may be assumed as its natural one—A. But A will be in equilibrium in any other position; for instance, in that shown by B, as the two forces still act in the same vertical line through *k*, as shown by *b b*. It is thus evident that such a substance or solid has no stability whatever.

Now the equilibrium can be made stable by shifting the point through which the centre of gravity acts. Assume that the specific gravity of the solid B, is made unequal, so that it becomes denser or heavier about *p* (see C); it is apparent that on such a change the *centre of gravity* would be shifted to some point, *g*, and the forces would no longer be acting in the same vertical line. The resultant of the buoyant pressure of the water would act upwards in the line *a a* through *k*; and the resultant of the weight of the body would act downwards in the line *b b* through *g*. The horizontal distance between the two lines *a a* and *b b* would be the couple upon which the two forces acted, until the solid got into the position D, where the two forces would act in the same vertical line *a a*. The equilibrium of a solid such as D floating would be stable, if, upon being inclined from its original position until in the position C, it had the power to regain the position D.

It has been proved that "form" of itself has no stability, and it remains to be shown how the variableness of form in a *partially* immersed body can create a stable condition of equilibrium. Let it be assumed that

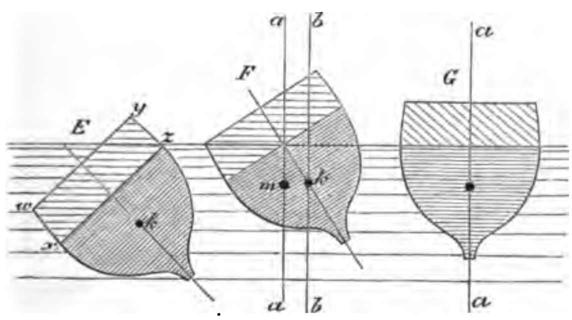


FIG. 18.

the solid A has an addition made to it, as illustrated in E (Fig. 18) by *w x y z*. The *bulk* will be increased, but the *weight* is to remain exactly the same, with the centre of gravity still at *k*. The body will rise in the water until in the position F, so that a part remains immersed still equal in *bulk* to A. Owing to the altered *form* of the immersed part of the solid,

the centre of buoyancy has shifted to some point *m*, but the centre of gravity remains at *k*. Now the resultant or buoyant pressure of the water in the line *a a* no longer acts through *k*, but through *m*, whilst the weight of the solid still acts through the centre of gravity, *k*, in the line *b b*. It is quite plain that the solid could not remain in the position F, but would take the original position of A, as shown by G, with the forces of buoyancy and gravity acting in the same vertical line *a a*.

The manner in which variations in form affects the stability of vessels can be illustrated in this way; if a vessel with such a section as that portrayed in Fig. 19 were filled out in the garboards at O O just above the

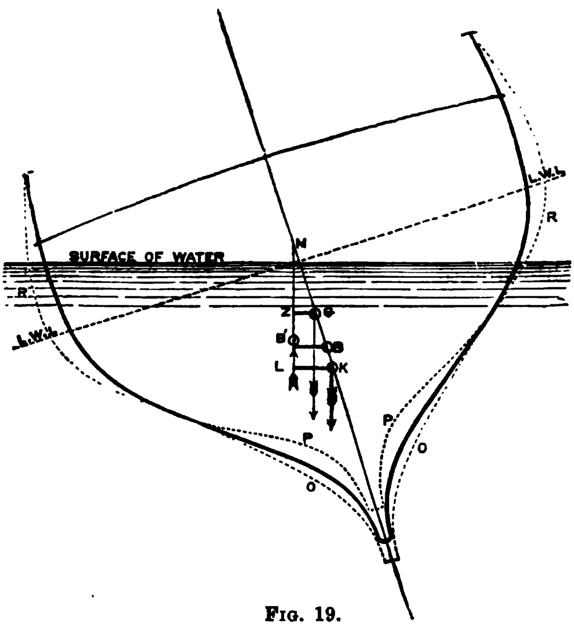


FIG. 19.

keel, it is plain that the centre of buoyancy (B) would be lowered, and the point M would be brought nearer the centre of gravity (G); therefore the arm of the righting lever G Z would be shortened; or in simple words the additional buoyancy placed at O O would be so much upsetting power. But in the case of a yacht the added displacement about O O would be utilised for the stowage of additional ballast; and by this means the centre of gravity (G) would be brought lower; so that it is quite possible that the original distance between G and Z would be maintained.

The effect of increasing the height of the centre of buoyancy relative to the surface of the water can be illustrated in this way. Assume that the displacement, or rather the hull, is cut away at the garboards as shown at

P P and added to the hull near the load water-line, as at R R, then the centre of buoyancy would be higher, and upon inclination of the vessel would shift out farther to leeward than shown by B′, so that the distance G Z would be increased, always supposing G to be kept in its original position by shifting the weights lower, such as could be done by putting additional weight on the keel. If the centre of gravity could be brought to K, and with the centre of buoyancy at B, the length of the righting lever would be K L. As a matter of fact, however, until the introduction of lead keels the centre of gravity was seldom found below the centre of buoyancy.

In considering stability it is necessary to understand its influence at extreme angles of heel to prevent capsizing; this influence is dependent upon what is termed the "range of stability."

By "range" is meant the angular distance through which a vessel can be heeled before she loses all righting power, or arrives at the vanishing point of her stability, when she would capsize.

The range will be better understood if graphically shown by a curve which shows the righting power at all angles of heel. (See Fig. 20.)

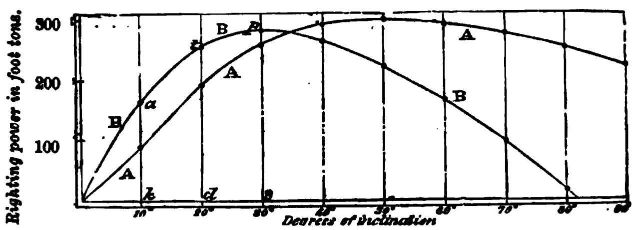


FIG. 20.

The distances *a k*, *t d*, *p s*, represent the righting power of a vessel, when heeled at various angles, as 19°, 20°, 30°, &c.; then the curved line *a* or B passing through the spots at the termination of the distances (as at *a*, *t*, *p*, &c.) are the "curves of stability." The curve A represents the curve of stability of an English yacht of the deep type, and it will be seen that the righting power is greatest when the vessel is heeled to 50°; this would be termed the vessel's maximum stability. The curve B represents the curve of stability of a vessel of equal length, but of greater beam, and much less depth and weight. It will be seen that this vessel has the greatest righting power at 30° inclination; this is the shallow vessel's maximum stability.

Now, it is this greater stiffness at initial angles of heel which, whilst it is of the utmost advantage for speed, forms the element of danger in shallow vessels. It can be supposed that a vessel, B, is sailing at an angle of 15°, and that a sudden acceleration of wind force heeled her

to 30°, the point where her maximum stability would be reached; then, if the wind force were not instantly removed the vessel would increase her heel if she got the least beyond the 30° until she finally lost all stability, or righting power, at 80° and capsized. But if a similar force were applied to a vessel with a curve of stability like A, when she reached 30° inclination there she would remain, and it would take a very large increase of wind to carry her to 50°, the point where her maximum stability would be reached; and even then there need be little danger of capsizing, as the decrease in the length of the righting lever is so slow, that at 90° there is nearly as much righting power as the shallow boat has at 30°. As a matter of fact, no wind force could lay a yacht, with such a curve of stability as represented by A, flat on her beam ends, inasmuch as long before she reached 90° the wind would have lost nearly all its effect upon the sails. The fact must, however, not be overlooked, that wind currents often come in a diagonal, or even in a vertical, direction; and a yacht might be hove down by such currents as these and fill, or the ballast shift so that she would not right; but such an event is a far-off contingency, and could not very well happen to a yacht with a crew on board to handle her. The curve A no doubt accurately represents the curve of stability of some broad and shallow American yachts prone to capsize. Such a yacht was the Mohawk schooner, which was capsized in a squall whilst at anchor off Staten Island in 1876, and the Grayling, which was capsized in a squall, whilst sailing off the wind in 1883. The latter schooner is 81ft. on the load-line, 23ft. beam, and 6ft. draught. After her deck became submerged, with main boom in the water, and head sheets flat in, she appears to have lost way and would not answer the helm whether it was put down or up; consequently she could neither be got into the wind nor before it. She continued to settle over very fast until on her beam ends, where she rested for twenty minutes, gradually sinking as the water forced its way through the hatches, skylights, &c. The Mohawk was a larger vessel than the Grayling, being 130ft. on the water-line, 30ft. beam, and 7ft. draught of water. The circumstances attending her capsizing also differ somewhat from those in the Grayling accident. The vessel was at anchor, and, no doubt, from this cause felt the force of the squall more severely. Her maximum stability would be reached say at 30°, and then the squall that put her over so far would take her to the vanishing point in less time than it takes to write it. But it is quite possible that the Mohawk would not have gone clean over on her beam ends if her ballast and heavy cabin furniture had not shifted, inasmuch as when she got to 60° the wind must have had decreased effect on the sails; and, as the squall passed over very quickly,

the Mohawk might have righted but for the reason stated. However, beyond the accident of the ballast and heavy furniture shifting, the Mohawk was subject to another condition, which rendered her righting quite impossible: she had a very large "well" or cockpit aft, into which the main cabin opened, and soon after the deck became immersed the water rushed into the cabin, and then, of course, her chance of righting was gone; she sank very suddenly, drowning her owner, his wife, and three other persons, all of whom were imprisoned below by the influx of the water. However, narrow or broad yachts with great draught of water and heavy lead keels, such as the Britannia, Iverna, Penitent, or other similar yachts, are in no danger, even if hove down on their beam ends, providing water is not allowed to get inside their hulls. The case, however, is very different with shallow vessels which have not deep lead keels and with some steamers, which have their centres of gravity high; and it is necessary for their safety that they should not be sailed very near the angle of heel where their maximum stability is reached. In practice, in small shallow boats, this is well understood, and the helmsman throws his little craft in the wind directly she is struck by a squall, or lets go sheet or halyards, whichever comes readiest to hand.

The manner in which form influences the range of stability will be shown farther on, illustrated by actual curves of stability of existing yachts; but it will be convenient here to refer to some experiments made with models by the author and Mr. E. H. Bentall to obtain a general knowledge of the effect of ballasting by lead keels on deep and narrow and broad and shallow yachts.

				Y.R.A. Tons.
No. 1 Model.	Length on W.L. ...	=	49ft.	= 5 Beams for length = 20
„ „	Extreme beam	=	9ft. 9⅝in.	
No. 2 Model.	Length on W.L. ...	=	55ft.	= 7 Beams for length = 18
„ „	Extreme beam	=	7ft. 10⅜in.	
No. 3 Model.	Length on W.L. ...	=	60ft.	= 9 Beams for length = 17
„ „	Extreme beam	=	6ft. 8in.	

In each Model (the freeboard) from W.L. to top of covering board = 3ft.
„ „ the depth from W.L. to bottom of keel .... = 10ft. 6in.
„ „ the length of heeling lever from W.L. .... = 33ft.
„ „ the top of ballast from W.L. .... = 7ft. 6in.
„ „ the displacement .... = 31 tons.
„ „ lead keel .... = 23·1 tons.

The models were heeled by weights on deck, but before giving the results of the experiments, it will be useful to describe their nature. Fig. 21 is a section of a vessel, and her centre of gravity must lie somewhere in the line V F M, which is the middle line of the upright position. M is the metacentre and F the centre of gravity (not centre of buoyancy). O is a weight on deck, which on being moved to one side, say to K

inclines the vessel, because the centre of gravity is necessarily moved when the weight is moved. The centre of gravity will move out to some point, at F^2, parallel to the line O K and less than O K as the weight shifted is less than the whole weight of the vessel. The line F^2 M is vertical, and the point of intersection at M shows the metacentric height. The manner of determining the point will be shown in the calculations further on.

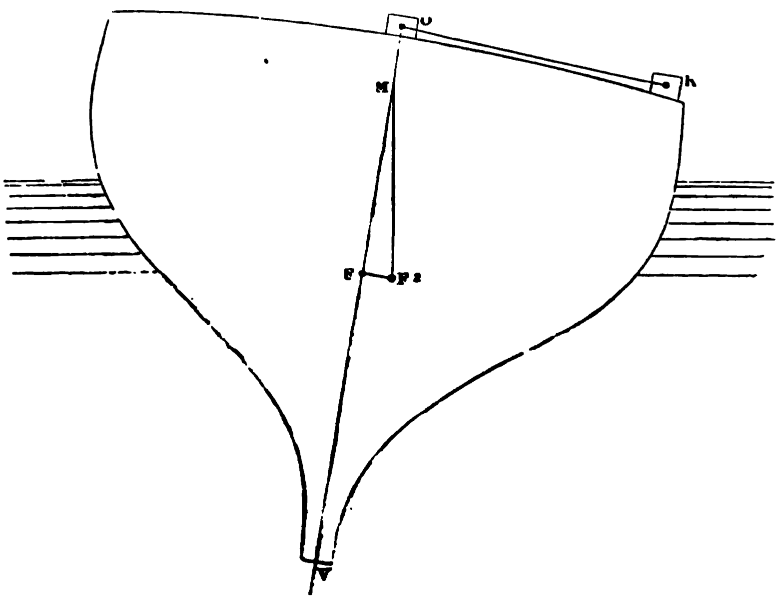


FIG. 21.

The experiments were conducted as follows: In Fig. 22 A is a perpendicular erected above the deck, from which a plumb line was suspended at A (the plumb line was a piece of thread and a small bullet); W is a weight of 5oz. (or 1·93 ton for a full-sized vessel). In the first experiment W was shifted 8ft. (by the scale $\frac{1}{2}$in. to a foot), and the plumb deflected from B to C 1-94ft. In the second experiment the weight was moved 12ft., and the plumb deflected 2·94ft. to D. The position of the centre of gravity of the yacht below the metacentre was then found by the formula $MG = \left(\frac{W \times k}{D}\right) \times$ cot. angle of heel.

MG = height of metacentre above the centre of gravity.

W = the weight moved.

k = the distance W was moved.

D = the displacement.

Cot. $= \frac{P}{Y}$ where P = the perpendicular height from the plank bearing the weight to A, the point of suspension of the plumb; and Y the horizontal distance the plumb line moved.

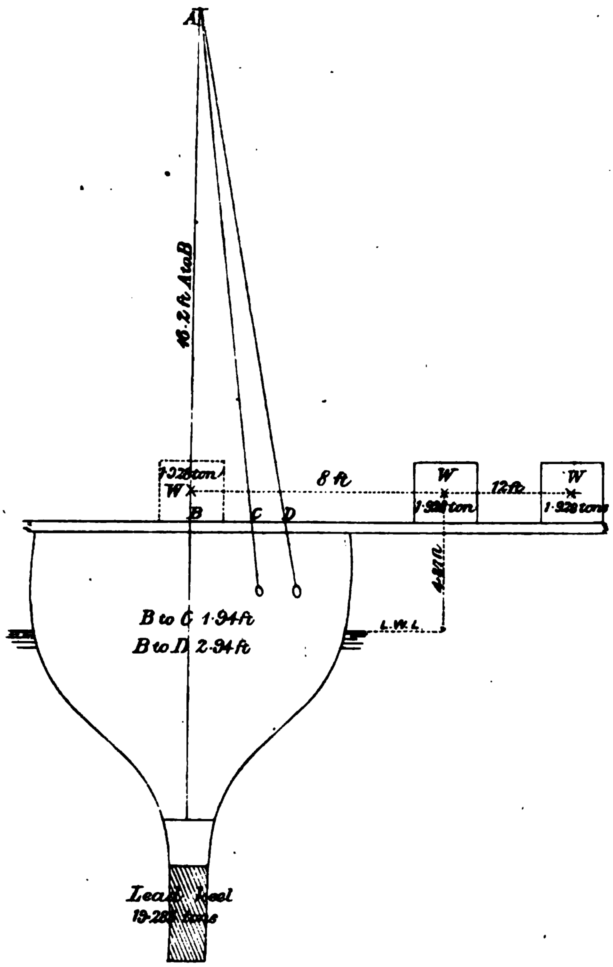


FIG. 22.

The calculations made from these experiments gave the following results :

No. 1 Model	First experiment,	MG = 4·235ft.
	Second „	MG = 4·206ft.
No. 2 Model	First experiment,	MG = 3·548ft.
	Second „	MG = 3·648ft.
No. 4 Model	First experiment,	MG = 2·921ft.
	Second „	MG = 3·113ft.

The height here given for MG is greater than would be found in actual yachts, as the spars, sails, heavy deck fittings, &c., were not included. However, for the purpose of comparison the results are reliable.

The chief feature disclosed by these results is that, taking the difference of the metacentric height between No. 1 and No. 3 models as 1·1ft., the weight of the ballast ought to have been lowered bodily 1·7ft to have retained an equal metacentric height, $\frac{1\cdot1\text{ft.} \times 31 \text{ tons D}}{20 \text{ tons ballast.}} = 1\cdot7\text{ft.}$

The breadth, it will be seen, had been reduced 3ft., and by lowering so much of the weight of displacement as was represented by the lead keel 1·5ft. or half the distance the beam had been reduced, equal stability could be maintained, the length being increased 11ft.

Assuming that the weights remained unaltered, the effect of reducing the beam and adding to the length on the sail-carrying power can be thus illustrated:

A 20-tonner would have 2000 square feet area of lower sail, and the centre of effort would be about 26ft. above the centre of lateral resistance. Lower sail could be carried in a strong breeze equal to a pressure of 2lb. per square foot. The heeling moment would be found by multiplying the sail area (S) by the wind pressure per square foot (P), and by the height of the centre of effort (H) above the centre of lateral resistance. $S \times P \times H =$ heeling moment in foot-pounds $= 104{,}000$.

The angle of heel at which the righting moment will balance the heeling moment will be thus found: $\frac{\text{Heeling moment}}{D \times MG} =$ sine of angle heel.* D (the displacement) is expressed in pounds, and, giving the quantities for No. 1 model, we have $\frac{104{,}000}{69{,}440\text{lb.} \times 4\cdot2\text{ft.}} = \cdot35$, which is the sine of 20° 30′, and the vessel with such a pressure would thus be heeled to 20° 30′.

With a similar wind pressure and sail area, No. 3 model would be taken to 30°, if ballasted the same as No. 1.

The effect of adding to the length without adding to breadth, depth, or displacement, was also considered. It was found that by adding 6ft. to No. 1 model, increasing her water-line length to 55ft., her metacentric height became 4·5ft. as against 4·2ft. with a water-line length of 49ft. The righting moment for this condition was found to be 109,368 as against 104,000 with 49ft. length of water-line. The effect of this would be that for the same heel another 100 square feet of canvas could be carried; or if the same sail area were carried the vessel would only be heeled to 19° 24′, and thus carry less canvas more effectually. From this it can be gathered that some advantage under certain conditions could be gained by increasing length, and examples of this in actual yachts will be given farther on.

To further carry on these experiments, the author made two models of

* The righting moment of any given angle of heel is thus found: D × MG × sine of angle of heel.

equal lengths, 49ft. The first was on the model of the Vanessa (Fig. 23), with 9·9ft. beam, and a lead keel 3ft. deep, bringing her draught to 10ft.

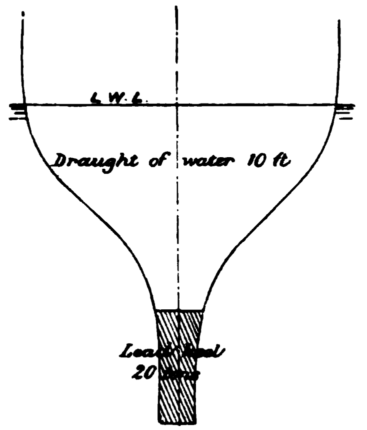


FIG. 23.

The weight of the keel (by $\frac{1}{4}$in. scale) was 20 tons, and the displacement of the hull 31 tons.

The second model was of three beams, or 49ft. long, 16ft. broad, and 3ft. draught of water to the rabbet of keel; displacement 31 tons, and ballast 20 tons.

The model of five beams was heeled four times, and the subsequent calculations did not show more than 0·1ft. variation in the metacentric height, the mean being 4ft. exactly, or 0·2ft. less than that recorded for Mr. Bentall's model of 6in. greater draught. Fig. 19 shows the midship section of this model.

The broad model was first heeled with the ballast inside, as shown in *Fig. 24.* Two inclinations gave exactly the same result, or a metacentric

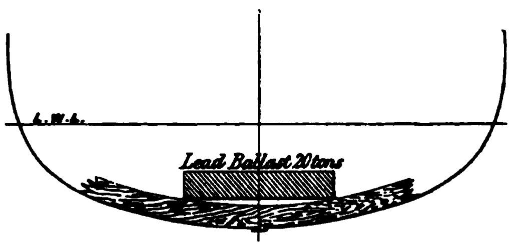


FIG. 24.

height of 4·8ft. (It might here be said that, for the sake of accuracy, a plumb line was used equal to 72ft. by scale, or 3ft. actually.)

This result showed that practically the broad model with her ballast inside, and the narrow model with her ballast outside, were equal in stability. The author has frequently pointed out that there is no special

virtue in the ballast being inside or outside, and its effect can only be measured by the effect it has on the metacentric height. In this case we find that by keeping the length constant, and reducing the beam 6ft., an equality can be maintained in stability by lowering a dead weight equal to two-thirds the displacement, by exactly the distance the beam

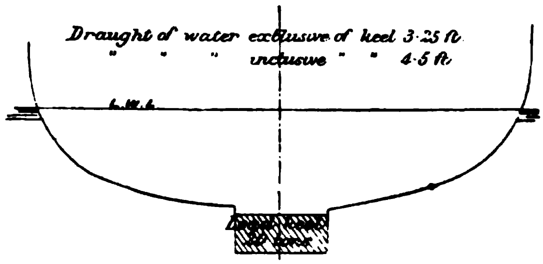


FIG. 25.

was reduced. In other words, the under-water depth was increased just as the beam was decreased. Of course, this uniformity would be destroyed if the weight shifted had been less than 20 tons, or if the weight of ballast carried by the two models had varied. This result was not unexpected, as it had previously been shown in the case of actual yachts.

The next experiment made with the broad model was by putting the lead outside, as shown by Fig. 25, making the draught of water 4·5ft. The height of MG for this disposition of the ballast was 6·8ft. The centre of

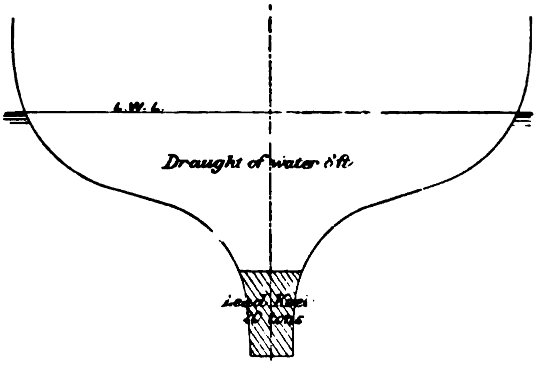


FIG. 26.

weight of the ballast was lowered 3ft., and a simple calculation showed that the heeling experiment gave a correct result.

Another model, of the same length, breadth, and displacement, but filled out in the garboards like a modern keel yacht (see Fig. 26), with twenty tons lead keel and 8ft. draught of water, was next heeled. Her height of MG

was found to be 8ft. The centre of gravity of the weight on the keel was lowered 2·25ft. The results of these experiments were as follows:

	Length. Ft.	Breadth. Ft.	Draught. Ft.	MG. Ft.
No. 1	49	9·9	10·0	4·0
„ 2	49	16·0	3·0	4·8
„ 3	49	16·0	4·5	6·8
„ 4	49	16·0	8·0	8·0

These results show the enormous initial stiffness* of vessels which are relatively broad and shallow, as, although each may be of the same weight, yet, owing to the different lengths of righting lever, one may have, as can be seen in the table above, just double the stiffness of another. Frequently, too, in practice, a vessel has less weight than another of the same length and yet has the same righting power; the Arrow and Kriemhilda are cases in point, as will be seen from the table here given.

	Thames tons.	MG Ft.	Displacement.
Lyra	364	4·00	332
Sappho	392	7·30	232
Sea Belle	142	3·32	155
Miranda	139	3·50†	160
Jullanar	126	3·30	158
Florinda	135	4·75	150
Rose of Devon	140	4.00	128
Arrow	115	3·35‡	106
Defender§ (American, 1895)	—	10·3	150
Kriemhilda	106	3·00	115
Carina (1894)	69	6·31	47·33
Luath	5	2·26	11·75
Olga	5	2·24	10·2
Trident	5	1·75	9·0
Clotilde	5	2·18	7·8
Currytush	3	2·05	7
Erne (S.Y.)	158	2·05	182

The Arrow, it will be seen, has a metacentric height of 3·35ft., and at 20° inclination the length of her righting lever would be 1·4ft. ($= 3{\cdot}35 \times \text{sine } 20^\circ = 3{\cdot}35 \times {\cdot}34 = 1{\cdot}14$), and 1·14 multiplied by her weight, 106 tons, gives the righting force at 20° in foot tons as 121.

The Kriemhilda has a metacentric height of 3ft., and therefore her righting lever at 20° would be $3 \times {\cdot}34 = 1{\cdot}02$. Her displacement is 115 tons, and $115 \times 1{\cdot}02 = 117$. Thus it will be seen that the lesser weight acting on a longer lever has more than an equal righting moment, and this might be a great advantage when sailing in smooth water owing to the smaller weight to drive, and the finer lines which could be obtained. A more notable case of the effect of

* Initial stiffness is the resistance a vessel makes to being heeled from the upright position.

† This is the MG of Miranda with 6 tons of lead on her keel. She has now about 30 tons.

‡ This is the MG of Arrow with about 14 tons of lead on her keel. She has now 40 tons and her MG is about 4·5ft., or nearly equal to that of a modern yacht like Valkyrie.

§ She was heeled without sails and gear on board, and her displacement at the time was estimated to be 150 tons.

metacentric height is shown in the case of Lyra and Sappho, the relative righting moment of each at 20° being Sappho 580, Lyra 441. Their relative lengths of righting lever at 20° are 2·5ft. and 1·33ft. as shown in Fig. 34, page 48. The case, however, might be wholly different if the yachts were sailing in disturbed water; the greater weight then might be an advantage, apart from stiffness altogether, as will be explained farther on.

Under the tonnage rule, as already shown (see page 13), it was necessary in order to make a successful racing yacht to do with as little beam as possible. It was, therefore, of paramount importance that the greatest beam should be at a point in the hull, where it would have the greatest effect on stability.

It is often erroneously supposed that stability is largely added to if the greatest beam is at the deck instead of on the load water-line, because

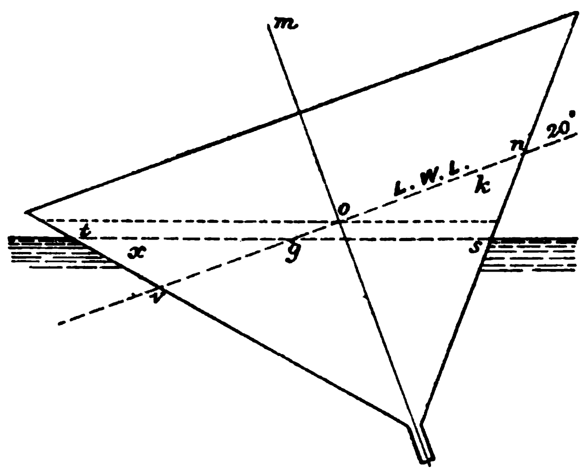


FIG. 27.

by putting a much larger piece in the water on the lee side than is taken out on the weather side, the centre of buoyancy is shifted farther out to leeward, and so lengthens the righting arm. It has already been pointed out that this condition cannot exist, and that the piece put in the water must necessarily equal in bulk that taken out.

If such a condition could exist then the triangular form of section, as shown by Fig. 27, would be the most powerful. But the case stands thus : Let L.W.L. be the water line of the upright position, then at an inclination to (say) 20° this line ought to intersect the middle vertical line *m* at *o*; but the fact is the vessel, as previously explained, would rise in the water until the wedges *k* and *x* were equal in volume, so that the point of intersection would really become somewhere at *g*.

This matter can be differently illustrated, if it is recollected that the

height of the metacentre above the centre of buoyancy is dependent upon the breadth and area of the plane of flotation and the under water bulk of the vessel; or the nearer the bulk of displacement is to the L.W.L. the higher will the metacentre be above the centre of buoyancy. If a vessel were inclined to 20° until the line *g s* (Fig. 27) became the half breadth of the L.W.L. on the emersed side, it is obvious that the total breadth of the plane of flotation at 20° inclination would be less than the breadth, *o n*, of the upright position, unless the breadth on the immersed side, *g t*, increased as much as that on the emersed side decreased. In other words, unless the whole breadth, *t s*, more than equalled the breadth, *v n*, there would be no gain in power; and if *t s* were less than *v n* there would be a loss of power.

But the clearest way to demonstrate the relative effect of the greatest beam being in one case at, and in the other above, the load line will

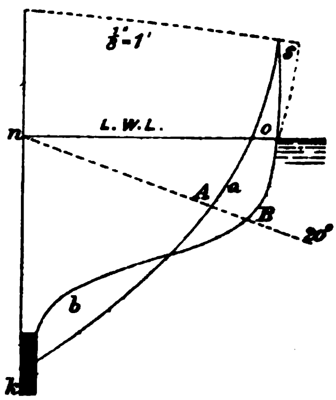


FIG. 28.

be shown as follows: Let A and B, Fig. 28, represent the midship sections of two yachts, each 86ft. in length, with an extreme beam of 17ft., and a displacement of 140 tons; the centre of gravity of each, regulated by ballasting, to be 2ft. below the L.W.L. The area of midship section in both is alike, the area *a* being equal to *b*.

In Fig. 29 the curve of B is shown, and it will be seen how greatly the righting power* exceeds that of A up to 65°.

A few years ago this might not have been the case, as the form of midship section of B (*see* Fig. 28) would not have admitted of the ballast being stowed so low in the hull as in A; now, however, that the whole of the lead is usually placed on the keel at *k* (Fig. 28), B can have her centre of gravity as low relatively to the L.W.L. as B, and her metacentric height will, of course, be greater.

* The cause and value of this power in actual yachts will be found illustrated on page 47.

This leads us to consider another aspect of the case; assuming that the greatest beam to be taken for a tonnage rating at the L.W.L., it would then clearly be some advantage to have a greater beam at the deck than that rated at the L.W.L.; or say A's beam (Fig. 28) were taken from *n* to *o*, then she would obtain some addition to stability by extending her beam to *s*; this is illustrated in the curve of B, whose beam is extended 2·2ft., as

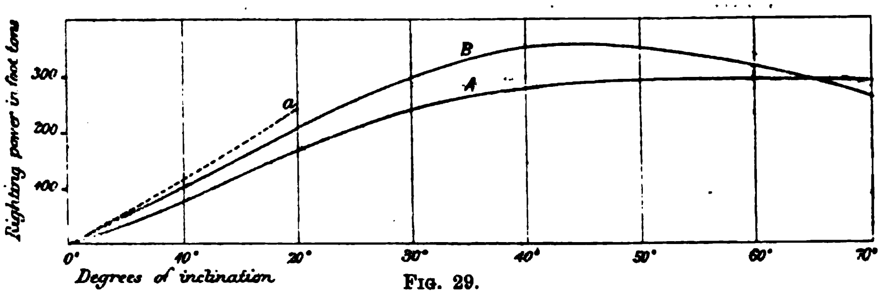


Fig. 29.

shown by the dotted line from *o* to *s*; the effect of this on her stability is shown by the dotted line *a*, Fig. 29. But, by the previous illustration, it is plain that if B were 22ft. broad at the load water-line instead of at the deck, she would be still stiffer in much the same ratio shown by the two curves A and B, Fig. 29.

From the foregoing it would appear that for any given beam its greatest breadth should be carried well down under water to get the best results, and should not turn to any considerable extent to shape the bilge until at a depth below the L.W.L. equal to the lowest free-board; or the bilge should not show out of water until an angle of 20° is reached. This was an essential condition for the narrow, deep yachts

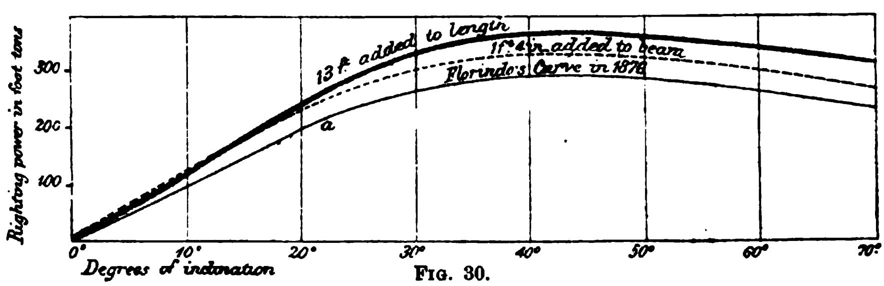


Fig. 30.

produced by the old tonnage rule, which heavily taxed beam; but now that the tax has been removed, a much freer use can be made of breadth, and the deep, hard bilge is no longer a necessity—in fact, it might be a great disadvantage on account of the wave making it induces.

To illustrate the relative effect of beam and length on stability in actual yachts, Florinda will be taken as an example. In Fig. 30 *a* is the actual curve of stability of the yawl Florinda as she sailed in 1876; the

dotted lines show what her curve would have been with 1ft. 4in. added to her beam, the centre of gravity remaining in the same position 2ft. below the L.W.L., and displacement being added to by 5 tons, bringing it up to 155 tons.

With 13ft. added to her length, her stability would be as shown by the dark curve, the displacement being added to by 25 tons, raising it to 175 tons.

This shows what the value would be of retaining the same vertical athwartship sections whilst lengthening out the interval between the sections: not only an increase of stability would result, but finer lines would be obtained; it should, however, be noted that 1ft. 5in. beam, with only a small increase in displacement, is equivalent to 13ft. of length with 25 tons increase of displacement. However, there is not the smallest doubt that the 13ft. of length would be of more value than the 1ft. 5in. of beam.

To further illustrate the relative influence, breadth, depth, and the locus of the centre of gravity have on the range of stability of yachts, a

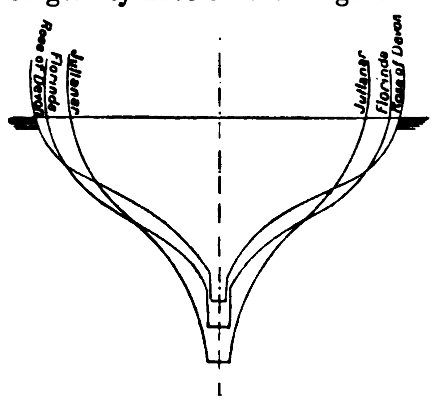


FIG. 31.

comparison will be made with the curves of the Rose of Devon, Jullanar, and Florinda, these vessels varying greatly in form, as shown by the sketch of their midship sections in Fig. 31.

	Rose of Devon	Florinda.	Jullanar.
Length on load line	81ft. 3in.	86ft. 10in.	99ft. 1in.
Breadth (extreme)	20ft. 8in.	19ft. 4in.	16ft. 10in.
Breadth on load line	20ft 4in.	19ft. 1in.	16ft. 8in.
Draught forward, 5ft. from stem	6ft. 8in.	7ft. 3in.	2ft. 9in.
Extreme draught	11ft. 9in.	11ft 9in.	13ft. 8in.
Area of load water plane	1120 sq.ft.	1100 sq.ft.	1085 sq.ft.
Area of mid section	95 sq.ft.	106 sq.ft.	106 sq.ft.
Displacement	128 tons.	150 tons.	158 tons.
Centre of buoyancy aft centre of length	1·2ft.	1·2ft.	0·4ft.
Centre of buoyancy below L.W.L.	2·6ft.	2·75ft.	3·44ft.
Metacentre above C.B.	6·1ft.	4·75ft.	1·13ft.
Metacentre above centre of gravity	4ft.	4·3ft.	3·3ft.
Area of lower sail	5220 sq.ft.	5257 sq.ft.	4988 sq.ft.
Ballast	57 tons.	54 tons.	79·5 tons.
Portion of this ballast on keel	none.	23 tons.	6 tons.

The Jullanar is (approximately) six times her beam in length; Florinda four and a half times, and Rose of Devon less than four times.

Their relative proportions of breadth and depth will be readily understood by the sketch of their midship sections.

It will be observed that Jullanar has much greater depth than either of the other two, and less beam; in fact, practically, the area of section which she loses by her contracted beam is made up by the excess quantity she has near the garboards.

The effect these differences in form have upon the stability of the three vessels can well be shown by curves. (Fig. 32.)

The Rose of Devon, it will be seen, has the greatest initial stability of the three; her curve is very steep to begin with, but her maximum stability

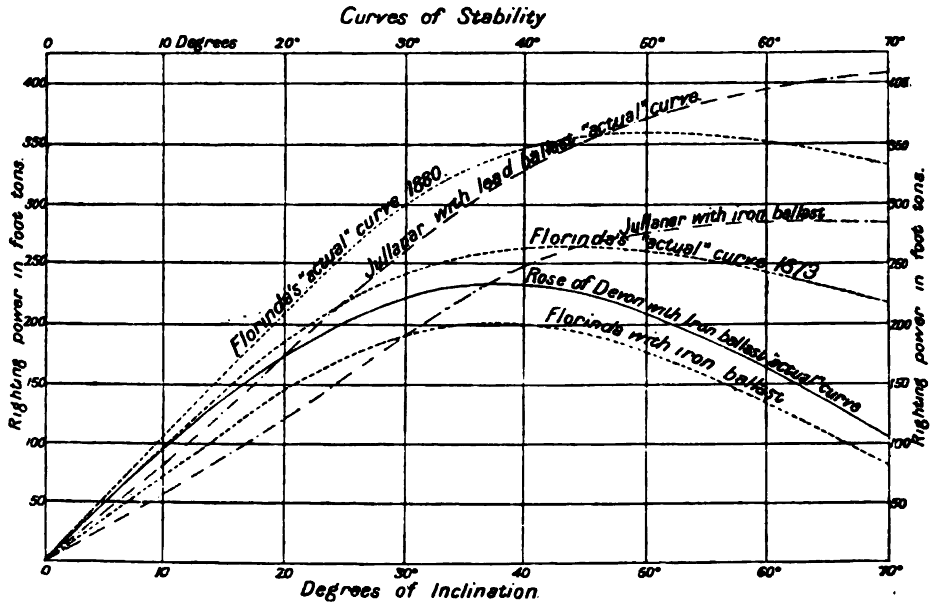


FIG. 32.

is reached at an angle of inclination of 40°, and then decreases very rapidly. In short, this curve can be taken as an example of the kind of stability a yacht has, which is broad in the beam but shallow in body.

The curve representing the Florinda is similar in character, but shows less power at small angles of heel. The Jullanar is a vessel quite distinct in type to either of the two named, and it will be seen that her curve is distinct also. She has very small stability at initial inclinations, but an almost unlimited range, and her greatest stiffness occurs at 75°, or when she is nearly flat on her beam ends.

To start with, her stiffness is very inferior to that of the other two, and she has to be heeled to 20° before she equals the Rose of Devon in

stability, and to beyond 25° before she equals Florinda, as the latter was sailed in 1873, but at 30° she exceeds them in righting power; at 40° their maximum stability is reached, but Jullanar continues to increase.

These three vessels exactly illustrate the truth of the proposition that additions to beam add largely to initial stability, and have no such effect in augmenting the range, whilst depth has such effect, and at large angles of heel does duty for beam. This can be better understood by a comparison of the areas of flotation of the three yachts. In the upright position these areas are—

Rose of Devon	1120 square feet.
Florinda	1100 „ „
Jullanar	1085 „ „

At an inclination of 50° these areas are found to be—

Rose of Devon	742 square feet.
Florinda	800 „ „
Jullanar	960 „ „

That is, the order of comparison is exactly reversed, and at 50° the Jullanar has 220 sq. ft. more area of flotation than Rose of Devon, whilst in the upright position Rose of Devon only had 34 sq. ft. more than Jullanar.

By referring to Fig. 33, the cause of the diminished area of flotation will be observed; in the broad shallow model the breadth of the vessel across the water-line has become reduced to *a b*, whilst that of the deep-

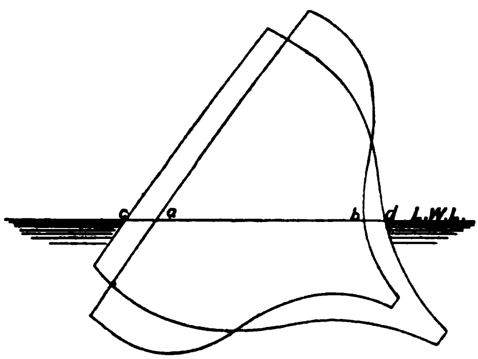


Fig. 33.

bodied, but narrow yacht, shows a considerable excess of breadth, as shown from *c* to *d*, in fact, when the narrow yacht is on her side, she is, as far as stability goes, somewhat in the condition of the broad yacht in her upright position. (*See* page 40.)

It will be noted that the Rose of Devon is ballasted with iron, and has no metal keel. Formerly this was a very common condition, and in order to compare her stability with the Florinda and Jullanar, which are ballasted with lead and have lead keels, their curves are given as if

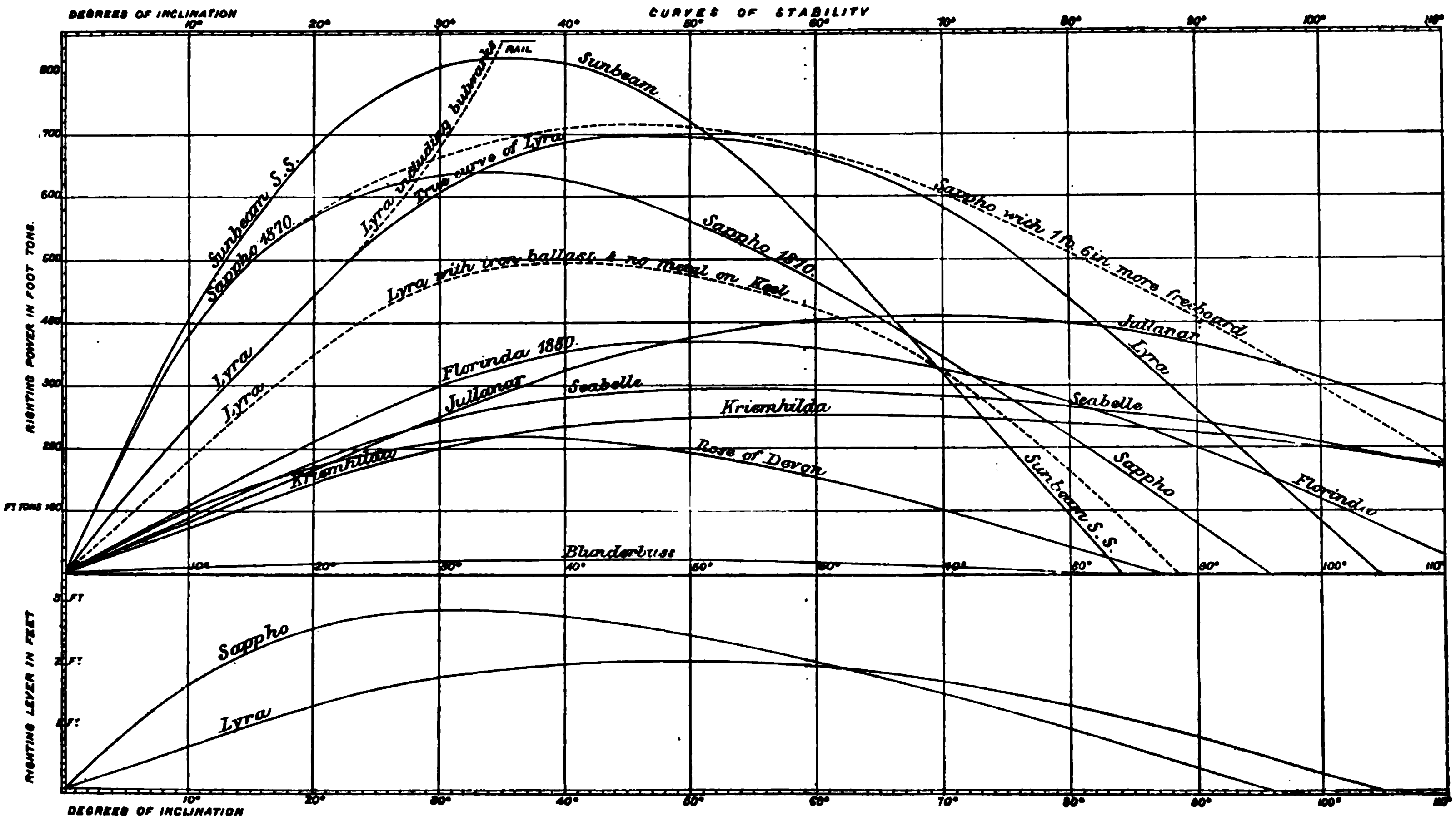

CURVES OF STABILITY
DEGREES OF INCLINATION
RIGHTING POWER IN FOOT TONS.
RIGHTING LEVER IN FEET
RAIL
Sunbeam
Sunbeam S. S.
Sappho 1870.
Lyra including bulwarks
True curve of Lyra
Lyra
Lyra with iron ballast & no metal on Keel
Sappho with 1ft. 6in. more freeboard
Florinda 1880.
Jullanar
Seabelle
Kriemhilda
Rose of Devon
Sappho
Florinda
Blunderbuss
Lyra

Fig. 34.

ballasted with iron as well as their actual curves. It should, however, be said that such a yacht as Jullanar would have been regarded as an impossibility in the days before lead ballast was introduced.

These curves show the exact relative stiffness of the three distinct types of yachts, and are applicable to the most modern types (1896); one

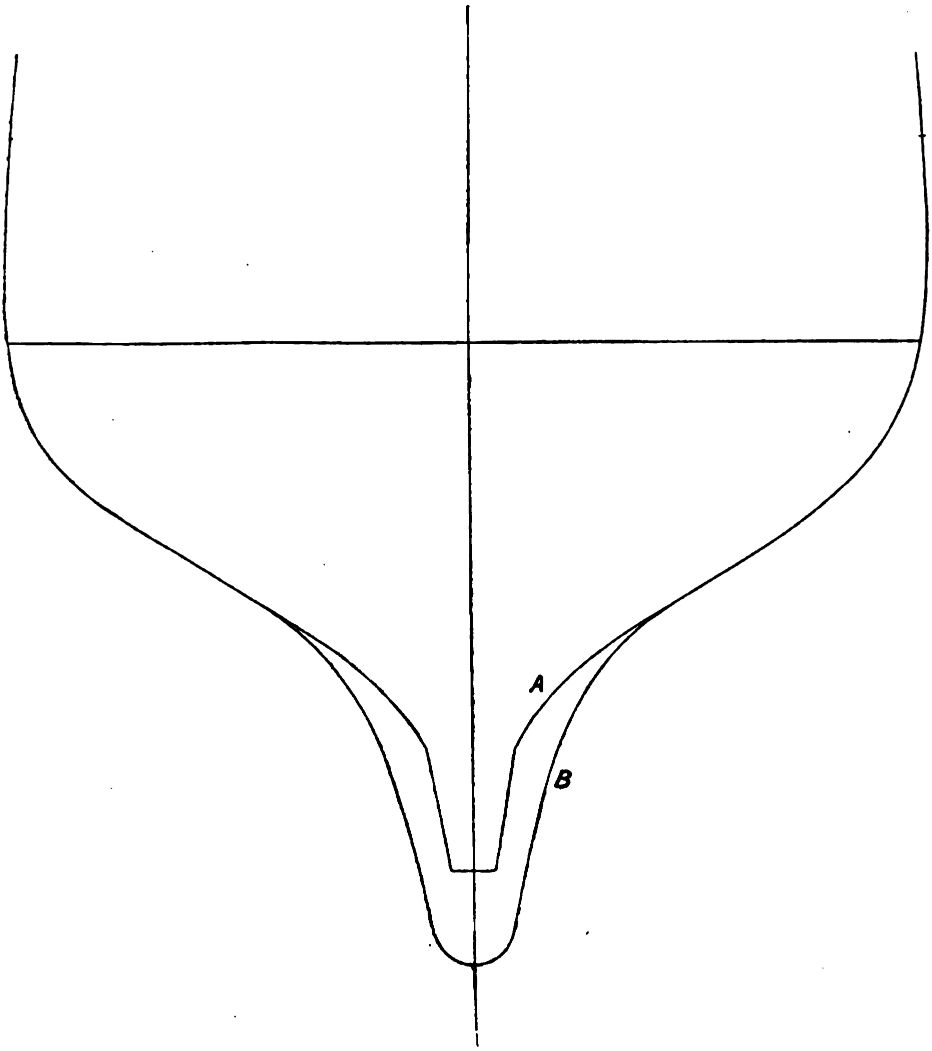


FIG. 35.

of them (Rose of Devon*) with iron ballast inside, but possessed of the greatest initial stability and the lowest range; and the Jullanar ballasted with lead, and showing the faintest initial stability and the longest range.

The general character of the stability of British yachts as influenced

* Rose of Devon has now a lead keel.

by the old tonnage rule for the purposes of competitive sailing has now been sufficiently illustrated; but it remains to be shown how the rating rule by sail area and length of water line affected the character of a yacht's stability. No yacht of the Jullanar type was built under the rule, and the yachts ranged from three to four beams in length; but the chief alteration in the outline of the mid-section was in the garboards and keel. This is illustrated by Fig. 35, which represents the Florinda's mid-section (four and a half beams to one), and what her mid-section would have been like had she been built as a cruiser in 1895. The garboard and keel portion A illustrates her section as it actually is, with 23 tons of lead on her keel. The outside line B shows what a modern Florinda is like with the whole of her lead (64 tons) on the keel. (*See* Fig. 31.)

The effect this alteration in the form of the lower part of the hull and in the manner of ballasting is shown in the curves of stability, Fig. 36.

Formerly it was a common plan to give a yacht a very low freeboard,

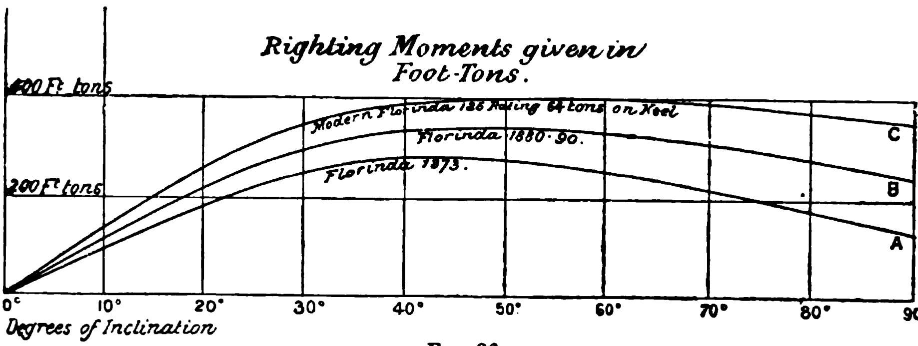


FIG. 36.

and to compensate for the deficiency by having high bulwarks. The object was to keep the deck and its weights as low as possible in order to lower the centre of gravity, and also to keep the centre of effort of the sails low. It was argued that the bulwarks would keep out the water until heeled to the rail, just the same as freeboard would, so far as sailing in a smooth sea goes; but experience showed that high bulwarks and a low freeboard form a very inconvenient and uncomfortable arrangement for sailing in a disturbed sea on account of the difficulty in getting the water off the deck. The effect that bulwarks have on lengthening the righting lever or increasing the stability after an angle which would put the deck under is reached, is shown by the curve of Lyra (Fig. 34), whereon the dotted line from 23° shows what the effect would be if her deck were raised to her rail. It must, however, clearly be understood that height of bulwark cannot be reckoned on as a permanent part of a yacht's stability, because directly the rail is under water the stability as suddenly diminishes

to that due to the deck edge limit. Stability, from a safety point of view, must therefore be treated in a manner wholly independent of the height of bulwarks.

The sort of curve a steam yacht has which is ballasted with lead and has a lead keel is shown by Sunbeam,* and the curve of a small steam yacht is exemplified by Blunderbuss.

* The curve of Sunbeam is taken from Sir W. H. White's valuable work on Naval Architecture.

CHAPTER IV.

THE PROPORTIONS OF YACHTS AND THEIR MOTIONS AMONG WAVES.

THE performance of a vessel in a disturbed sea is mainly governed by her length in relation to the length of the waves she encounters; by her depth under water, and height above water, relative to the depth and height of the waves; by her weight, relative to the weight or volume of the waves; and to the stowage of the vessel's weights in a fore and aft direction.

It is obvious that a vessel could be made so long and deep that Atlantic waves of ordinary length and height would be powerless to sensibly move her vertically or horizontally; that is, she would form a kind of floating breakwater—the waves breaking over her, but unable to impart motion to her of any kind. This condition can be readily realised by reflecting on the absolute stillness of a big Atlantic liner when lying in a comparatively sheltered anchorage like Spithead in a strong wind travelling at the rate of twenty-five miles an hour; whilst the wave disturbance is quite sufficient to cause a vessel of, say, 100 tons to pitch and 'scend and roll in a lively fashion, whether she be lying at anchor or under way.

The old theory was that a vessel to be a good sea boat must have great breadth of beam; and of this there formerly was no doubt, as a narrow vessel would, owing to the ballasting adopted, be deficient in stiffness, and consequently wallow in the sea in such a manner that sailing her would have been out of the question. Now, however, with our knowledge of what can be done with depth and lead ballast, stiffness can be left out of consideration, and attention wholly devoted to the attainment of as much length as may seem desirable for any given tonnage in order to secure a boat which is slow and easy in her fore and aft motions.

In 1851 a cruising or racing yacht of about 100 Thames tons would have had a L.W.L. depth of about three and a half beams, or 70ft. long, 20ft. broad, with a depth of hold of about 8·5ft. amidships, 10ft. extreme

draught, and a displacement of about 120 tons; but the operation of the Thames rule and Y.R.A. rule of measurement gradually brought into existence a yacht 32 per cent. longer, 20 per cent. narrower, and 40 per cent. deeper; that is, a yacht of 100 tons in the year 1886 would, in length on the L.W.L., exceed five and half beams, or be about 90ft. long on the L.W.L. and 16ft. broad, with a depth of hold of about 13·5ft., and an extreme draught of water of about 14ft., and displacement of 175 tons. The different manner two such vessels would perform in the short steep channel sea, such as would be met with anywhere round our coast in a strong wind, would be most marked. The short, broad, and comparatively shallow vessel would fall into every trough, and lift over every crest, whilst her vertical motion would be excessive. The longer vessel, on the other hand, if judiciously canvassed, would avoid many of the wave hollows, and never have her head thrown up by a wave crest with the suddenness, nor to the extent, which the shorter vessel would; and, although her speed through the water would be much greater, her increased weight would very sensibly diminish the power of the waves to cause vertical motion. It might at first be supposed that the longer, narrower, and deeper yacht would be a much wetter vessel, but this would be by no means the case provided she was of proper stiffness, so that she did not wallow in the sea; but the fact must not be overlooked that the longer boat is not the better and faster sea boat so much because she is narrower but because she is longer and deeper, and of greater weight or displacement.

The last sentence must be circumstantially impressed on the reader; the longer yacht of "100 tons" did not perform better in a sea because she was *narrower* than the 70-feet yacht; but because she was 20ft. longer, 5ft. deeper, and 50 tons heavier; in short, because she was a much larger vessel. Both yachts were described as of "100 tons" by the rule of measurement, and this empirical manner of comparing yachts engendered a belief, or we might almost term it a delusion, that a narrow yacht, or say one with a length of five and a half beams, was superior or easier and faster in a sea way than one of four and a half beams. Persons who did not give the matter much thought were easily convinced that the narrow yacht had come into existence because narrowness best suited our seas; but the fact is, the delusive way of estimating tonnage or size wholly masked the merits of broader yachts of actual equal size or cubical contents. A yacht, for instance, 90ft. on the water line, 20ft. breadth, and 13ft. depth of hold, and 175 tons displacement, would be a better vessel in every way than one of 90ft. length, 16ft. breadth, and 14ft. depth, and 175 tons displacement; but because the yacht of 70ft. length and 20ft. beam, and 110 tons displacement, and the yacht of 90ft. length, 16ft. beam, and 175 tons were

indiscriminately made 150 tons, the longer yacht was by comparison regarded as the better vessel; so also because by the rule the vessel 90ft. by 16ft. was made of 100 tons, whilst the 90ft. yacht of 20ft beam was made 150 tons, the narrower yacht was still regarded as the better vessel.

A modern racing yacht of 90ft. length has from 23ft. to 25ft. beam, and her weight may range from 150 to 160 tons, and, taken altogether, she is a most wonderfully efficient vessel in a sea way on any point of sailing. She is able to carry an enormous sail spread, by virtue of her great beam

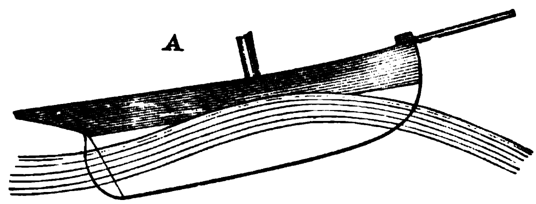


FIG. 37.

and the low position of her lead keel, the underside of which in some cases is so much as 18ft. below the water surface, whereas 12ft. was considered quite sufficient in the days of a 90ft. narrow yacht. The latter type would not, however, have the smallest chance against one of the modern 90ft. yachts in any kind of weather. It should, however, be pointed out that the introduction of girth in the rating rule has already had the effect of reducing the draught of water, but it is unlikely that the coming yachts will be, on the whole, any better or easier sea boats than those built during the decade between 1886 and 1896.

It does not necessarily follow that because a yacht is large or long or long and narrow that she may not be an uneasy sea boat, and we will now

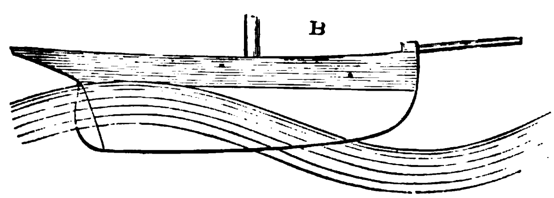


FIG. 38.

briefly consider the conditions which influence the motions which are generally described as pitching, 'scending, and rolling. Pitching is the motion akin to diving which a vessel takes after her head has been thrown up by a wave as A, Fig. 37, which is 'scending, and she then passes through that wave as at B, and falls down into the trough of another wave, C, which is the pitching motion as shown in Figs. 37, 38, and 39, the wave crest *d* being ready to break on board before the vessel recovers herself.

These fore and aft motions are influenced by the form of the vessel at and about the water-line; thus if a vessel is very fine on the load water-

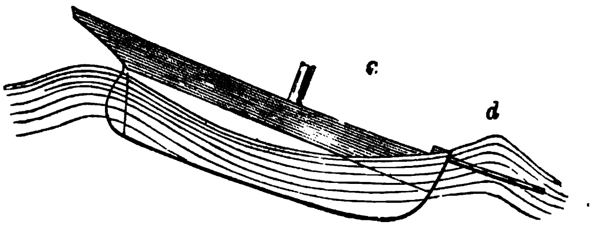


FIG. 39.

line forward and full aft, her pitching motion may be much more marked than if she were fine both fore and aft—a notable example of the latter form will be found in the fore and aft lines of Jullânar. It is obvious that

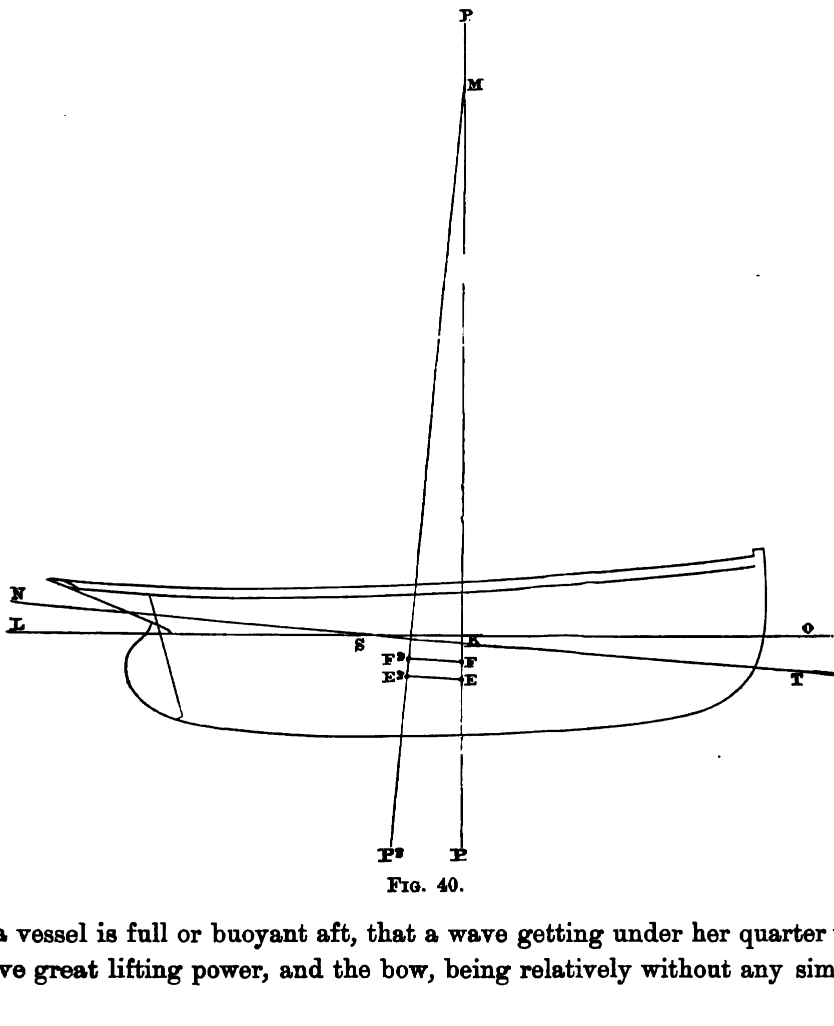


FIG. 40.

if a vessel is full or buoyant aft, that a wave getting under her quarter will have great lifting power, and the bow, being relatively without any similar

support, will necessarily be "dived" under, and a vertical motion will take place which is thus measured by what is known as the longitudinal metacentre. In Fig. 40, let L O be the normal load water-line, E the centre of buoyancy, and F the centre of gravity, P P produced through both. If by altering the immersion fore and aft by trimming, or by waves, the centre of buoyancy* might become transferred to E^2, the new load line being N T; the longitudinal metacentre for the position would be at the point M P^2 at right angles to N T.

It can be further assumed that the centre of gravity of the load water plane is at same point far aft at S; then the vertical motion is effected in proportion to the distance S K and the angular extent of the motion. The vessel has her fore and aft motion about an axis at S, the centre of gravity of the load water plane, and a consequent rising and falling of the vessel takes place each time she pitches or 'scends, and the extent of this vertical motion is thus quantified; $\overline{S K} \times$ sine angle of pitching or 'scending.

When a vessel is heeled as she usually is when under sail, the pitching or 'scending motion may be increased according as the centre of gravity of

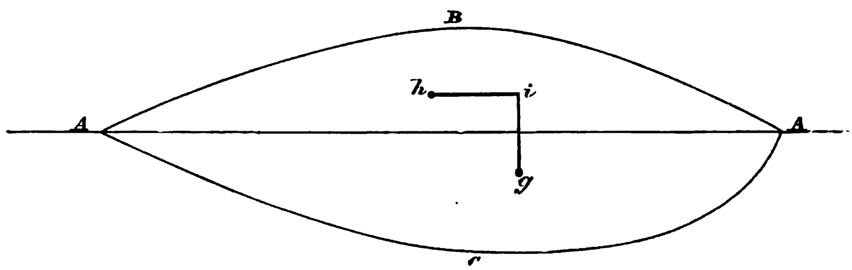


FIG. 41.

the immersed wedge is $\binom{\text{abaft}}{\text{ahead}}$ of the centre of gravity of the emersed wedge. In most cases the centre of the immersed wedge is abaft that of the emersed, and the consequence of this condition can be thus measured. In Fig. 41 let A A be the fore and aft middle line of the vessel, and A B A a plane in the emersed wedge, and A C A a plane in the immersed wedge; *g* the centre of gravity of the immersed wedge, and *h* the centre of gravity of the emersed. Through *g* produce *g i* at right angles to A A, and through *h* produce *h i* parallel to A A; then *h i* is the length of a coupling lever tending to depress the bow and lift the stern.

In finding the pitching moment for an inclined position of the vessel, the couple *h i* of a plane would be resolved into S (Fig. 40), as the *whole* plane would be treated as the plane of flotation of an inclined position

* F^2 in the diagram (Fig. 40, page 55) is merely put to show that if the centre of gravity were shifted to that point, to alter the trim, the centre of buoyancy would necessarily shift aft also.

and therefore the centre of gravity of that plane would be found. This would be readily done by multiplying the distance the centre of gravity of each plane—A C A, A B A—is from one end of the vessel by its area; the moments thus found would be added together; the sum divided by the *whole* area, would, in the quotient, give the distance the centre of gravity of the inclined plane is from that end of the vessel about which the moments were calculated.

It is obvious, from the above considerations, that if the centre of gravity of the immersed wedge were at h, and that of the emersed at g, the forces would be tending to depress the stern and lift the bow; and the extent of this depression could be calculated by a process analogous to that for the depression by the bow.

The centres of gravity of the immersed and emersed wedges of Sea Belle (calculated at an inclination of 20°),* are not in the same transverse plane; in fact, the centre of gravity of the immersed wedge is 1·26ft. abaft the centre of the emersed wedge. The mean volume of the wedges is 721 cubic feet; then by inclination of the vessel the centre of buoyancy would be shifted aft in proportion to the volume of the wedge, the distance their centres are apart, and the volume of displacement, thus:

$$\frac{721 \times 1{\cdot}26}{5425} = 0{\cdot}167\text{ft.} = \text{the distance the centre of buoyancy would be shifted aft.}$$

But, inasmuch as the centre of gravity of the vessel would not be shifted, this movement of the centre of buoyancy would not take place; on the contrary, the vessel would go down by the head until the centre of buoyancy came to rest in the vertical in which rested the centre of gravity. The extent of this depression by the bow can thus be approximately computed. The half length on the load line is divided by the longitudinal metacentric height above the centre of buoyancy, and the quotient multiplied by the distance the centre of buoyancy has to be shifted; thus:

$$\frac{\text{L M}}{\text{L W L}} \times {\cdot}167 = \frac{45}{81} \times {\cdot}162 = .09268\text{ft.}$$

Thus Sea Belle, at 20° inclination, will practically have her plane of flotation in the horizontal of the plane of flotation of the upright position, as her bow would only be depressed 1⅛in. However, in some vessels the centres of the wedges are separated to a much greater extent, and this fact, coupled with the condition that the centre of flotation of the upright position being far abaft the vertical in which the centre of buoyancy is found, will tend to increase the extent and violence of pitching. But,

* The Sea Belle is introduced here because exact calculations respecting that yacht are given farther on.

apart from pitching, it is obvious that if the bow is much depressed through a vessel being inclined, the form of her entrance will be greatly altered. Moreover, the centre of the longitudinal section (centre of lateral resistance) will be carried forward, and this will tend to make the vessel gripe or carry weather helm. According to Professor Rankine, the difference between the centres of the wedges of immersion and emersion does not exceed ·003 of the breadth in ships that are easy among waves as to pitching; on the other hand, he declares that in uneasy ships it is sometimes as much as ·04 of the breadth. In Sea Belle the difference between the centres is ·065 of the breadth. In the case of Kriemhilda the difference is as much as ·09 of the breadth.

From these considerations, it will be seen that the easiest and driest vessel will be one whose various centres are in the same vertical-transverse plane, and that the separation of the centres tends to aggregate pitching and uneasy motion generally.

The violence of pitching and 'scending is much affected by the radius of gyration. The radius of gyration is computed by multiplying the square of the distance each weight of different parts of the vessel is from their common centre of gravity; the products so found are added together, and the sum, divided by the whole weight of the ship, gives the radius of gyration about the centre of gravity. The more the weights are concentrated, as in modern yachts, the shorter the radius of gyration will be, and the vessel will consequently acquire less momentum in pitching, and will pitch and 'scend with the waves; but, on the other hand, if the weights are lengthened out, the momentum acquired by the motion of the vessel when her bow or stern is suddenly deprived of support during the passage of waves, will be much increased. By reducing this momentum, the "diving" and violent part of the pitching will be much modified, and what may be termed the "longitudinal rolling" will occur more with the time of the waves.

Lead ballast, whether stowed inside or outside in the form of a keel, has proved of the utmost value in placing the radius of gyration under the control of the designer, as generally when a yacht was ballasted with inside iron it had to be stowed very far fore and aft on account of the room it occupied; whereas lead by its smaller bulk, weight for weight, can be much concentrated, and at the same time occupy a lower level in the vessel. On the other hand, if the ballast be too much concentrated, the radius of gyration will be so much shortened that the vessel, whilst losing her diving tendencies, will be much too lively in disturbed water, and, whilst keeping time with the waves, rising regularly to every crest, would "jump all the wind out of the sails." This is particularly noticeable

in small vessels or boats, and to make the sails sit quietly whilst sailing among waves, the steersman must be very watchful to keep the sails full.

These lively motions are, however, easy, and not nearly so disagreeable as the violent diving plunges of a badly ballasted yacht in a heavy sea; in fact, for real comfort and dry sailing the ballast cannot very well be too much concentrated, and for good all round advantages it has been found that the dead weight should extend no farther than over the middle third of the length on the load water line.

The fore and aft motions of a vessel are influenced somewhat by the shape of the cross sections of vessels, and generally the easiest vessels are those which carry their bilges low down, so that there is little difference in

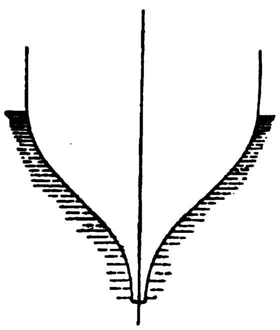


Fig. 42.

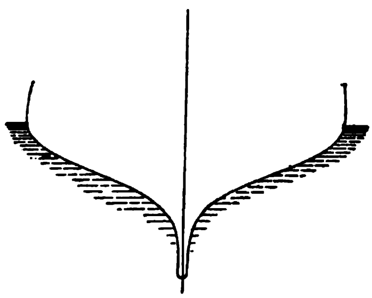


Fig. 43.

bulk in the immersed and emersed wedges upon heeling. Vessels with sections like Fig. 42 would be comparatively easy, whilst those approaching Fig. 43 are most affected by wave motion.

This is particularly apparent in transverse motions which come under the head of rolling; but to properly understand the conditions which entail easy, uneasy, safe, or unsafe, rolling, it will be best to first have a clear conception of what takes place during a vessel's oscillation, or rolling, in still water.

In Fig. 44, F is the centre of gravity of the vessel, and E the centre of buoyancy. Upon inclination of the vessel, the centre of buoyancy shifts to some point E^2; then the measure of the righting lever tending to bring back the vessel to the upright position, after being rolled to an angle shown by L.W.L. and the water level, will be the horizontal distance between the vertical lines passing through E^2 and through F. In proportion to the length of this righting lever, or the metacentric height, with certain limitations hereafter to be considered, will be the length of time the vessel will occupy in performing her oscillations.

The point about which a vessel oscillates is termed the instantaneous axis, and is dependent upon her form and position of the centre of gravity. It can thus be found: In Fig 44, the point S, where the water-line of the inclined position and that of the upright position intersect, would be treated as a tangential point in a curve, such as *x x*; the point of intersection of any number of water-lines for a corresponding number of different inclinations, would give other points for *s* in *x x*. The curve, *x x*, resting, assumed to be resting, or oscillating on the plane of the water, will

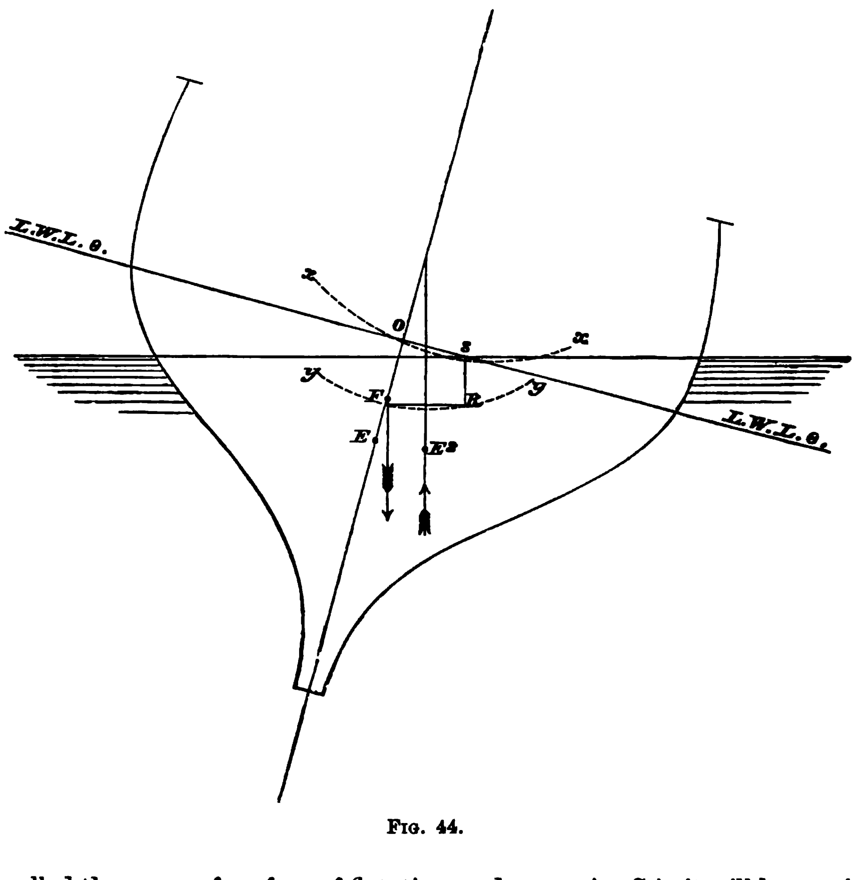


FIG. 44.

be called the curve of surface of flotation, and any point S in it will be a point for determining the instantaneous axis of the vessel. The centre of gravity will lift with the vessel in a vertical direction; hence, the instantaneous axis will lie in the horizontal distance F R, at its point of intersection with a vertical line through S. Thus, R will be the instantaneous axis, and may be found anywhere in such a curve as *y y*.

If the vessel were cylindrical in form, the point S would be at O for all inclinations; but, inasmuch as the wedge-shaped piece put into the

water is in excess of the piece taken out, a corresponding lifting of the vessel takes place, until the "in" wedge equals the "out."

This lifting of the vessel (called the lifting of her centre of gravity) has great influence on a vessel's oscillations, and produces that rising and sinking which are a part of uneasy rolling. This cause of uneasy rolling cannot well be entirely avoided in either yachts or ships, but it will be most observable in such vessels as flare out greatly above the water-line and turn in very quickly below, in the way some broad and shallow craft do.*

But easy rolling, or its converse, is to a great extent governed by the period which a vessel occupies in performing a complete oscillation. This period is understood as the time she occupies in making a double roll. The longer this period can be made, the easier and slower or less violent the rolling will be; but it is frequently found that the means available for lengthening the period can only be used at the sacrifice of some other and more important quality necessary to the good behaviour of the vessel. For instance, the period is largely dependent upon the depth the centre of gravity of the vessel is below the metacentre, and a vessel's period can always be lengthened by bringing the centre of gravity and metacentre closer together. It is plain that this could only be done at the cost of some stability, and this contingency would require very serious consideration if the vessel to be treated happened to be a racing yacht; and, moreover, the very stiffness which caused the yacht to roll quickly and uneasily when nearly before the wind, would tend to keep her steady, by admitting of a great press of canvas being carried, when she had the wind abeam or forward of the beam. Thus, so far as the uneasy rolling of racing yachts is influenced by a low centre of gravity, no remedy is admissible.

A vessel's period of rolling can, however, be governed to a small extent by her transverse radius of gyration; and, indeed, her "period" is calculated from her transverse radius of gyration and from her metacentric height. However, the calculation of the transverse radius of gyration would be much too complex a matter to venture upon, and the extent of the radius is generally deduced from an experimental roll after the ship is afloat. For the purpose of yachts it will be sufficient to know that the longer the radius the longer will be the period of rolling, and the radius can always be lengthened by transversely spreading out whatever movable weights the vessel carries. This, as a rule, could not well be effected by the ballast, as the latter is generally stowed in so narrow

* It should be noted that it is these lifting motions which cause the most disagreeable sensations in seasickness.

a space that to "wing" it out would involve its being lifted several feet; and such an alteration in the stowage of the ballast would, of course, seriously affect a vessel's stability. However, when a vessel is likely to be a considerable time with the wind aft, or on the quarter in a sea, the rolling may be slightly influenced by stowing her spare sails, chains, anchors, or gear in her sides or "wings."

The pitching which sometimes accompanies rolling, is an uneasy twisting motion largely influenced by the relative position of her centres of gravity of the immersed and emersed wedges. This influence has already been explained, and all we need point out here is that every effort should be made to bring those centres as near together as practicable.

Although yachts, owing to their great metacentric height, their form,

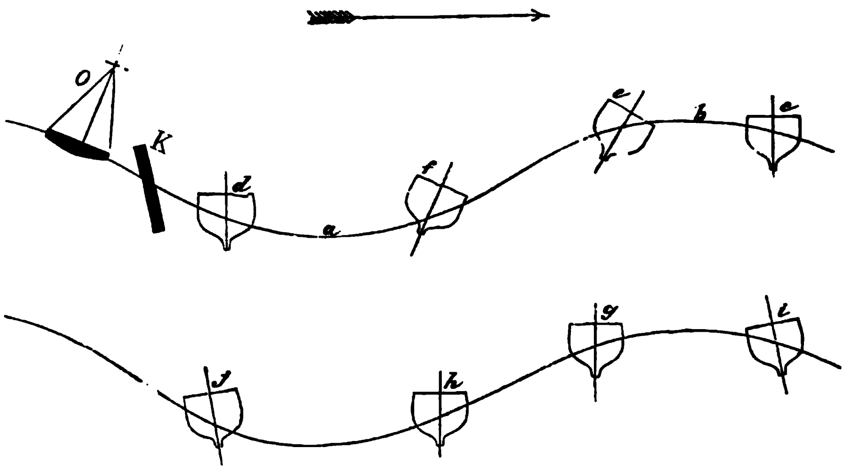


FIG. 45.

and the length of their masts (the latter lengthening out the vertical radius or gyration), are more or less subject to uneasy rolling, yet in some respects their form below water is conducive to easy rolling. In Fig. 45, let O be the transverse section of some vessel, such as a raft; such a vessel would have great initial stability, and she would practically always lie on the surface of large waves (as shown in the figure), with a tendency to roll with the slope of the waves. This would be the action of such a vessel on very large waves whose "period" much exceeded her own. But her "period" would be very limited, and if she got among small waves of her own period her rolling would be violent, "jerky," and dangerous.

Next take the case of a body such as K, of no stability, floating edgeways in the water; such a form, instead of keeping normal to the wave's surface, will retain the position it had before the water was formed into

waves. Supposing the wave to be travelling in the direction of the arrow, its tendency would be to turn the raft over in the direction of its own motion or slope; whereas the influence will have an exactly opposite direction on K, and, in fact, be tending to turn her against the slope of the wave. These contrary tendencies, explained by the fact that if a body such as K were oscillating large bodies of water would be set in motion, will exemplify the influence a keel, great rise of floor, dead wood, bilge keels or centre boards have on steadiness when a ship is among waves where she would roll considerably, for it is plain that if K were fixed underneath O, its effects would be to prevent O rolling so far with the wave. Thus, the influence of a keel is to retard rolling; that is to say, it would prevent her rolling to so great an angle as she otherwise would, and, at the same time, render her time for completing a roll longer; thus it would tend to produce steadiness. It should be remembered that a vessel's "period" or time consumed in making a double roll is practically the same, whether she oscillates through an arc of 40° or 10°.

From what has been said, it will be seen that the tendency a flat-floored vessel has to roll is very much checked by the presence of her keel, but that keel, if weighted, or if composed of iron or lead, would (whilst still opposed by the resistance of the water to its own motion) be a means of accelerating the vessel's oscillations, as it would lower her centre of gravity, and thus shorten her "period." The bad effect of lead or iron on the keel of flat-floored shallow vessels, under conditions which render a certain extent of rolling, with a following sea, a quarter sea, or beam sea, is well known. By lessening the vessel's "period" (through increasing the metric centric height) her rolling becomes more rapid or violent; or, for the sake of illustration, say the range of her rolling is 20° from the vertical each way, equal to an arc of 40°; she may go through that arc in five seconds without a metal keel, and through the same arc in three seconds after one has been fitted. On the other hand a heavy metal keel may be desirable to give a vessel a necessary amount of stability; this is especially the case in steam yachts, and it is a satisfaction to know that if such a metal keel is used in conjunction with bilge keels, the result will be to enhance the apparent effect of the latter. It should also be noted that placing a portion of a vessel's ballast below her centre of gravity would tend to lengthen her radius of gyration, and this in a small degree would lessen the bad effect of increasing the metric centre height. Great objection has been made to bilge keels on the grounds that they injuriously affect speed, steering, &c. These objections are only true of bilge keels when they are too long or not fitted parallel to the motion of the water disturbed by the ship; and in the case of properly arranged bilge

keels the effect on resistance or speed is almost entirely frictional, just the same as that of an ordinary keel which occupies the fore and aft centre line of a vessel.

It has been said that the period of a ship's rolling should exceed the period of the waves (that is, the time or period the wave takes to travel its own length), and there can be no doubt as to the truth of this. If the period of the unresisted rolling of the ship in smooth water be similar to the wave time in which she is placed, she would ultimately turn completely over, as the extent of her roll would constantly increase in proportion to the slope of the waves. But ships are so formed that the period of this unresisted rolling is always extended. "Unresisted rolling" supposes that no water will be set in motion by the oscillations of the ship, but, as a matter of fact, a very large body of water is set in motion by the keel, the dead wood, and the skin resistance each time a vessel oscillates, the effect of which will be to modify the rolling or extend the period; and, moreover, waves of uniform length, and consequent uniform period, are never met with; so the danger spoken of, arising from infinitely increased rolling, can be regarded as a theoretical one. The most that can be said is, that the vessel whose unresisted time of oscillation agrees with that of the waves she is amongst will always have her extreme angle of roll limited in proportion to the keel and other resistances.

There is no doubt, however, that at some time or the other a vessel will get among waves of her own period; these will be readily understood upon studying the table of lengths of waves and their periods at the end of this chapter. Every one who is accustomed to the sea will have noticed that a vessel when among waves of a certain form and length will roll violently; and yet when among some larger or perhaps smaller waves the rolling will be comparatively easy. In waves say of 40ft. length or under a shallow, stiff vessel of say 20ft. beam would roll very little; whereas a vessel of 10ft. beam would roll excessively. On the other hand the stiffer vessel might be the worst roller in waves of 80ft. length.

It follows that a ship rolling among waves will have an upright position, and in the case of such forms as O and K (Fig. 45) these upright positions would occur in the trough, *a*, or on the crest, *b*, of the wave. In the case of O, her great stability would keep her normal to the waves, whilst K, conforming to the motion of the water, moving from left to right in the direction of the arrow, would incline against it as before explained. The case of K is equal to that of a vessel whose unresisted period to the wave period is as $\sqrt{2} : 1$. Now, if the period of the vessel is not equal to this value, she will have her vertical positions before the trough or crest of the wave reaches her at *c* or *d*, and her greatest angle of roll will exceed

that of the slope of the wave, and occur after the trough or crest has passed her at *e* or *f*; so that there would be danger of the crest of a wave breaking aboard. If, on the other hand, the period of the vessel exceeds the proportion stated, the vertical position will occur at *g*, *h*, after the crest or trough of the wave has passed her, and consequently the angle of her greatest roll will not equal the steepest slope of the wave as shown at *i*, *j*.

To illustrate the danger a vessel might be subject to when carrying canvas in a beam sea, it can be presumed that the vessel's maximum stability is reached at an inclination of 20°, and that it vanishes altogether

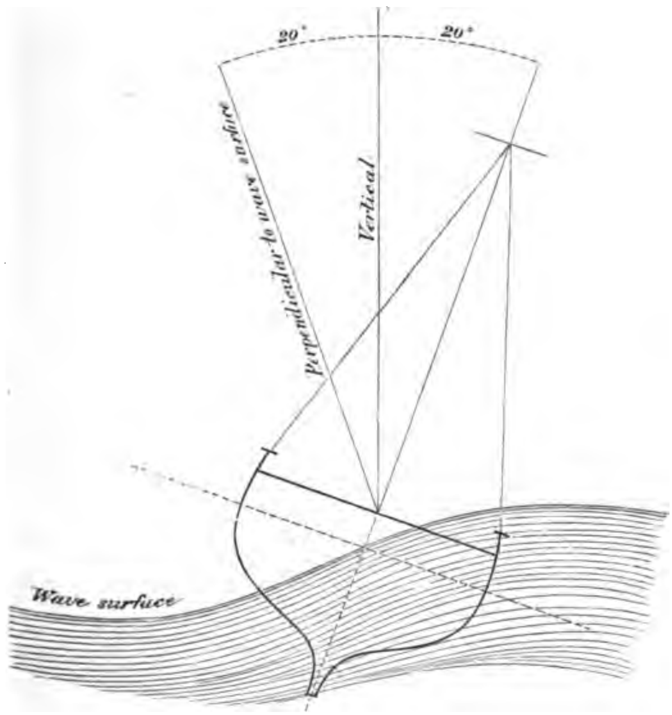


FIG. 46.

at 40°. Then, if the vessel were carrying canvas which would keep her to a permanent heel of 10°, any sudden application of that force after a lull would carry her to 20°, or a sudden gust might do so; if, also, her greatest angle of roll equalled the greatest slope of the wave (assumed to be 20°), and if this roll took place at a time when the vessel was struck by the squall, it is plain that her heel would, relative to the slope of the wave, equal 40°, and she would, in fact, upset. (*See* Fig. 46.) However, so far as English yachts are concerned, this is a purely hypothetical case, and the illustration is merely given to point out a danger that a vessel or open boat with a very limited range of stability may be subjected to.

It has been explained that yachts have a tendency to roll with the waves, like the raft O, and that, owing to their keel and dead wood resistance, illustrated by K, they also have a tendency to resist rolling with the waves. But the effect of K on the keel cannot be regarded as the result of an active force such as that of the stability of a vessel, which causes her to keep normal to the wave's surface, or to regain an upright position after being heeled; the keel, by being moved against the water, moderates or retards the heeling or rolling of the vessel only in proportion to its surface and the velocity with which it is moved; and by the same reasoning it retards the vessel regaining an upright position after being heeled or rolled. Thus, a deep keel, or centre board, or great rise of floor, will tend to modify the excessive rolling due to the vessel's period equalling that of the waves, and will generally have a steadying effect on a vessel when rolling broadside on to the waves; but it must be clearly understood that a deep keel or centre board would not prevent a vessel heeling under the influence of a wind pressure on her canvas; it would cause the process of heeling to be slower, but nothing more; unless in the case of a centre plate it were made heavier than water—of iron or lead, for instance—then lowering it would lower the centre of gravity, and so add to the general stability of the vessel. If made of wood, or of a material lighter than water, lowering it would still tend to retard the rolling and render the process of heeling slower; but it would also somewhat lower the centre of buoyancy of the vessel, and thus in a small measure decrease her stability.

Frequently, a yacht's tendency to "yaw" is much more troublesome and dangerous than her uneasy oscillation. This tendency is most marked when the vessel is running with the wind more or less on the quarter; and if the steering be not skilful, she may either "come to" so much as to be in danger of broaching to, or fall off so as to be in peril of gybing, either of which operations might be attended with very serious consequences. A vessel is most likely to "come to" or come nearer the wind when going down the slope of a wave and near the trough, and fall off when near or rising to the crest of a wave. The tendency to yaw will be most pronounced in a vessel which is very full aft, or which has a very raking sternpost and "rockered" keel, and whose heel and forefoot are consequently much rounded away. A large quantity of dead wood aft or drag will, on the whole, modify yawing; and a much rockered keel or very great rake of sternpost should be avoided if steadiness on a course is desired.

For the full mathematical consideration of rolling reference can be made to the researches of Mr. Froude, Mr. Rankine, and Mr. Scott Russell,

published in the Transaction of the Institution of Naval Architects, 1861, 1862, 1863, 1864, 1875; Naval Science, 1873, 1874, 1875; and to Sir W. H. White's "Naval Architecture."

TABLE OF LENGTHS OF WAVES, THEIR VELOCITY AND PERIOD.

Length in Feet.	Velocity per Hour. In Knots.	Velocity. Ft. per Second.	Periods in Seconds.	Length in Feet.	Velocity per Hour. In Knots.	Velocity. Ft. per Second.	Periods in Seconds.
0·56	1	1·688	0·33	143·8	16	27·01	5·26
2·25	2	3·376	0·66	162·3	17	28·70	5·59
5·06	3	5·064	0·98	182·0	18	30·38	5·92
9·00	4	6·752	1·31	202·8	19	32·07	6·25
14·05	5	8·44	1·64	224·7	20	33·76	6·58
20·2	6	10·13	1·97	247·8	21	35·45	6·91
27·5	7	11·82	2·30	272·0	22	37·14	7·24
36·0	8	13·50	2·63	297·3	23	38·82	7·57
45·5	9	15·19	2·96	323·6	24	40·51	7·90
56·2	10	16·88	3·32	351·2	25	42·20	8·23
68·0	11	18·57	3·66	379·8	26	43·89	8·56
80·9	12	20·26	3·99	409·6	27	45·58	8·89
95·0	13	21·94	4·30	440·5	28	47·26	9·21
110·1	14	23·63	4·60	472·5	29	48·95	9·54
126·4	15	25·32	4·93	505·7	30	50·64	9·87

The height of waves to some extent depends upon their length, and also on the depth of water, the waves made in shallow water being comparatively short, steep, and deep. The table which follows has been compiled from data found in Sir W. H. White's well-known Manual of Naval Architecture.

Lengths of Waves (179 observations).	Height of Waves.		
	Maximum.	Minimum.	Average.
100ft. and under	20ft.	3·3ft.	6ft.
100ft. to 200ft.	22ft.	5·0ft.	10ft.
200ft. to 300ft.	30ft.	7·5ft.	12ft.
300ft. to 400ft.	24ft.	10·0ft.	15ft.
400ft. to 500ft.	33ft.	12·5ft.	21ft.
500ft. to 600ft.	34ft.	16·2ft.	28ft.

The height of ordinary waves appear to range between one-twentieth and one-thirtieth of their lengths.

CHAPTER V.

CENTRE OF LATERAL RESISTANCE.

LATERAL resistance is the resistance the water offers to a vessel being moved sideways or laterally; this lateral or broadside motion when a vessel is underway is usually termed leeway, and is at right angles, or nearly at right angles, to her forward or direct motion; thus the resistance to leeway is properly described as "lateral resistance." The centre of lateral resistance is usually understood to mean the centre of the vertical longitudinal section of the immersed portion of the vessel, including the rudder. In other words, this immersed longitudinal section is assumed to be a plane; and if this plane be moved through the water in a direction at right angles to its own (the plane's) surface, then the resultant of the resistance it will meet with will act through its centre. For instance, let Fig. 47 be the immersed longitudinal section of a vessel with its

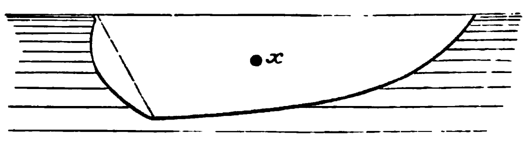


FIG. 47.

centre at x. If a towing line were attached to the point x, the vessel or plane would be towed laterally or "broadside on" through the water, without exhibiting any tendency to turn one way or the other; in fact, the plane representing the longitudinal section of the vessel would keep normal, or at right angles to the towing line. But if the towing line be attached farther aft, then on being towed the stern would come round towards the line; or if attached farther forward, the bow would turn round towards the line.*

* A simple experiment conducted as indicated with any model and piece of string will determine the centre of lateral resistance, disregarding any variation in actual position which might be due to forward motion, and which motion would not be given to the model during the broadside towing. The centre can also be found by the practical method described farther on among the various rules for determining the qualities of a vessel.

In calculating the centre of lateral resistance of a ship or yacht, it is always assumed that a plane has to be dealt with, and the immersed vertical-longitudinal section is taken as that plane. As a matter of fact, the centre of this plane would not be the centre through which the resultant pressure on the side of the ship would act. Owing to the varying form of a ship or yacht, it is almost impossible to determine by calculation the point through which the resultant of the *horizontal* pressure of the water actually acts; and, moreover, if the exact point could be readily determined, the knowledge of it would be of small practical value, for the reason that, owing to the motion of the vessel ahead in a line with her keel, there is an excess of pressure on the bow and a constantly decreasing pressure towards the stern; the bow is continually entering "solid" water, whilst towards the stern the water becomes more and more disturbed; and beyond this there will be an accumulation of water rising on the lee bow which has the effect of altering the form of the immersed portion of the vessel, and of adding to the pressure; this of itself carries the centre farther forward. And further, even supposing the centre of pressure could be accurately calculated for the upright position, it would be useless for any other position of the vessel, as a different portion of her hull would be immersed, or its position relative to the horizon altered, each time the vessel rolled or heeled.

For practical purposes, however, it is found useful to treat the centre of the plane as if it were the true centre of lateral resistance, as will be hereafter shown in dealing with the centre of effort of the sails.

Fig. 48 is the midship section of a yacht inclined to 20°, and the large arrows indicate the direction the resistance offered by the water to the horizontal or broadside motion of the vessel. It will be seen that the upper part or immersed side of the vessel presents a very effective surface for resistance; but it is not so with the lower part, between the bilge *a* and the garboard *b*. From a mathematical consideration of the resistance of fluids, we are taught that if *a b*, representing a portion of the hull (Fig. 48), is a plane moving in a fluid in the horizontal direction *b c*, then no more particles meet the plane than will meet the perpendicular *a c*; and therefore the resistance is diminished as *a b* to *a c*, or as 1 to sine of angle made by *a b* to direction of motion.

The other portions of the hull between *a* and the water surface would be treated in the same way, also the keel represented by *b d*; thus for the keel *e c* will be the effective surface for lateral resistance. Now assume that the straight portion of the curve *a b* had been continued in a straight line to *f*, it is evident that *a c* would be increased to *a g*; and in like ratio *e c* would

be decreased to *e g*. So far there would seem to be no advantage gained by carrying the section in a straight line to *f*, and it is pretty certain that in the angle at *f* some dead water would be carried along, and on the whole, the surface for frictional resistance to forward motion would be greater with an angulated garboard than with a curved one. There yet is another pressure at work, the result of the reaction of the horizontal pressure on *a b*, or *b d*; this resultant accumulates vertical pressure shown by the *little* arrows; these upward pressures (which will be minus on the weather side) whilst increasing the ardency of the lateral resistance will add somewhat to the

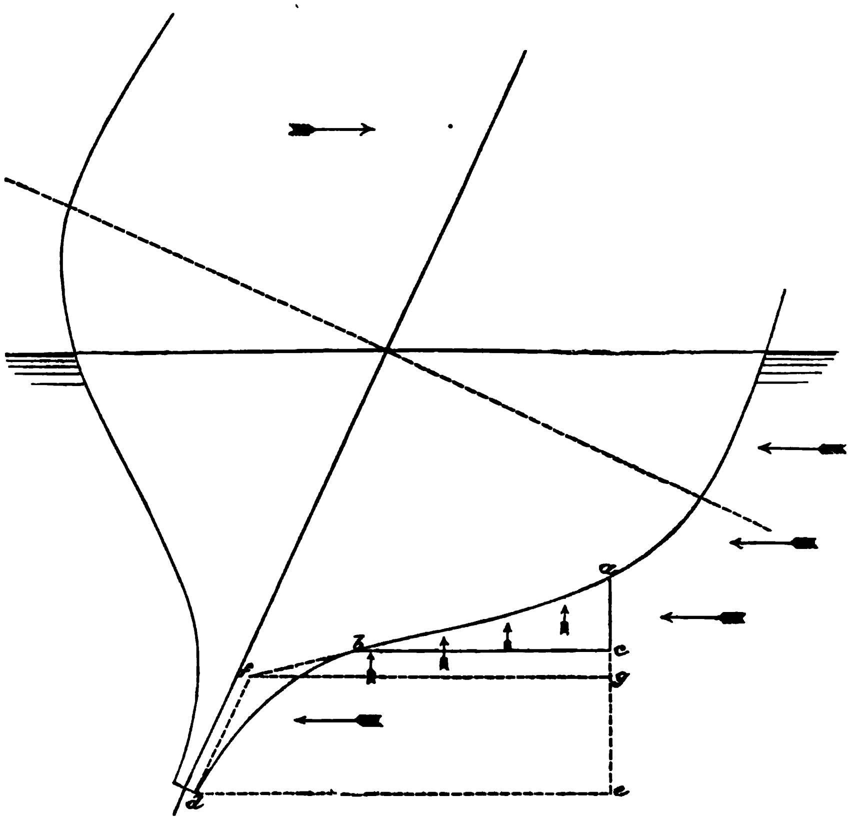


FIG. 48.

stability, and the flatter the floor the more considerable the addition will be.

It can be seen that a flat surface (such as the dead wood aft the keel, and the flat sides of the entrance) is more effective for lateral resistance than a concave or convex one; and this fact partly explains why it is that a boat fitted with a centre-board, as a rule, makes less leeway than a fixed keel boat in proportion to the total area of their vertical-longitudinal plane. It is true that in a flat floored centre-boarder *a c*

would be still shorter at 25° inclination; but then a centre-board boat is not usually sailed to an inclination exceeding 15°, so the pressure on *d f* (representing the board and dead wood), being more nearly at right angles to the direction of motion, would thus be intensified. It should, however, be stated that the lateral resistance due to friction, although small would be augmented by a flat floor or the concave surface of a hollow garboard.

If it were desired to attempt an accurate computation of the resistance

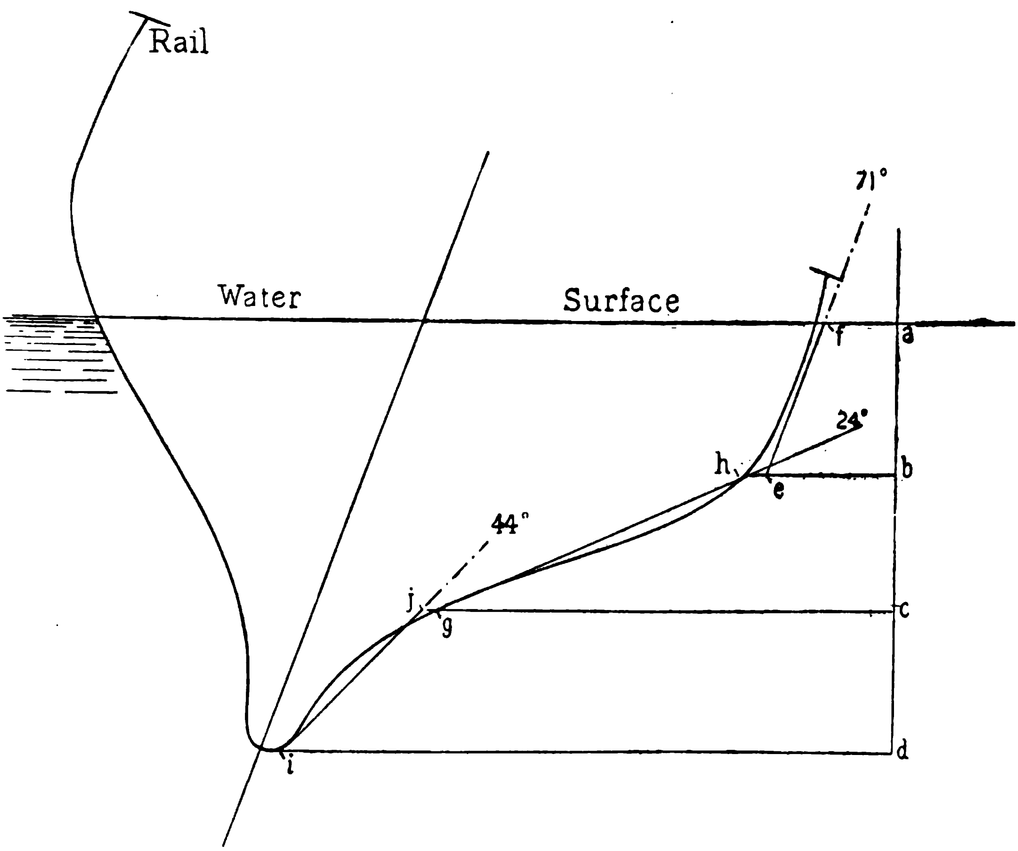


FIG. 49.

of the surface of the hull in a lateral direction the various cross sections at a given angle of heel, or in the upright position, could be sub-divided at equal vertical distances as represented by the horizontal lines *a, b, c, d,* drawn parallel to the water surface (Fig. 49 which shows a method adopted by Mr. Benjamin Martel). The average angle of the section with the horizontal lines as *b e f, c g h,* and *d i j,* would then be measured and the pressure on each angular surface, as on *e f*, say, computed

by the factors given in the table for the approximate law* of variation of

	10°.	20°.	30°.	40°.	50°.	60°.	70°.	80°.	90°.
Approximate Law	·220	·440	·640	·800	·900	·930	·960	·980	1·000
Law of Sines	·174	·342	·500	·643	·766	·866	·940	985	1·000
Law of Square of Sines....	·030	·117	·250	·413	·587	·750	·884	·970	1·000

pressure on oblique surfaces. It would, however, be quite safe to take the perpendicular *a d* as the sum of the pressures, *a d* being nearly the average sine of the various angles between *f* and *i*. A method for estimating the effective surface for lateral resistance for any given angle of heel would be to find *a d* for each section in the body plan at that given angle of heel, and substitute *a d* for the ordinates used in the example for calculating lateral resistance given later on; but, as a means of comparing the effective area for lateral resistance one yacht with another, the upright position can be used unless it is desired to see the difference in lateral resistance between one yacht heeled say to 15° and another to 20°.

It will be gathered that lateral resistance will always be greatly governed by the stability of the vessel, for it is plain that the more upright a vessel can be kept, the greater distance *a d* (Fig. 49) will be, and, in short, more effective will be the whole surface of the vessel to resist lateral motion.

The various lateral actions of the water would have to be more or less differentiated by the forward motion of the vessel, the ultimate effect being that the broadside motion or leeway would be decreased as the speed increased and also the resultant pressure of the water would be carried forward in proportion to the speed. The resultant pressure of the wind on the sails would also be carried forward, but not to the degree that the centre of lateral resistance would be, owing to the accumulation of pressure on the lee bow. Thus a large area of flat surface or dead wood aft is necessary, and whilst it can be more easily obtained in vessels with fine after bodies, it is in them least required, inasmuch as the water about the sterns of such vessels would be less disturbed than it would be in the case of vessels with full after bodies.

In reference to the pressure being carried forward it must be considered that the upper part of the dead wood aft, owing to the disturbed water it passes through, meets with less pressure than does the dead wood forward, as the bow is always entering new or undisturbed water; and, moreover, as before explained, there is usually a greater accumulation or

* For a great many years it was assumed that the pressure on oblique surfaces varied as the square of the sine of inclination; but the experiments of Beaufoy, Hutton, Vince, and Froude show that the true pressure varies more nearly as the simple sine, the true law being very near (although not absolutely determined) that given in the table as the approximate law.

rise of water at the bow. Hence "drag," or a much greater draught aft than forward, has been found of great use in keeping the centre of lateral resistance in a required distance aft, as the lower parts of what may be termed a raking keel are continually being moved into solid or undisturbed water. This matter can be illustrated in this way:—In the diagram (Fig. 50) let A be an immersed plane moving in the direction of the arrow *s*; and let it be assumed that the plane has also a sideway or lateral motion, as indicated by *t*. Next, *k* and *a* are points or spots on the plane, and *x* and x^1 are particles of water. As the plane moves forward and glides past *x* and x^1, the spots *k* and *a* will push them severally on one side, it being always assumed that the plane has sideway motion, and it is resistance to this sideway motion which we are considering. When any other indefinitely near spots on the plane, as *b h*, arrive abreast of *x* and x^1, they find the latter receding, in consequence of the push they received from

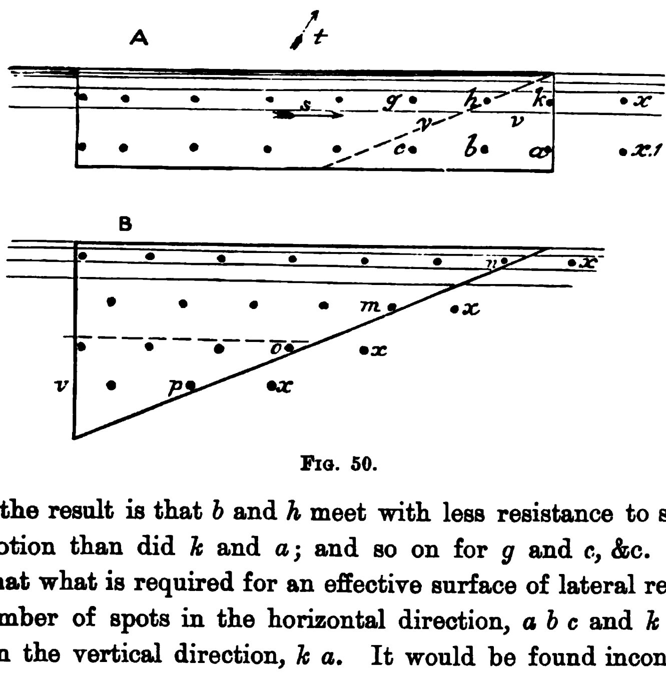


FIG. 50.

k and *a*; the result is that *b* and *h* meet with less resistance to sideway or lateral motion than did *k* and *a*; and so on for *g* and *c*, &c. It is thus evident that what is required for an effective surface of lateral resistance is not a number of spots in the horizontal direction, *a b c* and *k h g*, but a number in the vertical direction, *k a*. It would be found inconvenient to so increase the depth at the fore end to obtain an effective surface, but fortunately it is found to be an advantage to have an increase in depth at the aft end. Assume a triangular piece *v* to be cut off the fore-end of the plane A, and to be placed underneath aft as shown by *v* in the Diagram B; the area of the surface remains exactly the same, but a double number of spots as *n m o* and *p* are obtained that will enter solid water to meet particles of water as *x x*, &c., as the plane advances. It is quite patent that, although of equal surface, the plane B would more greatly resist

lateral motion if attended by a simultaneous forward motion, than would the plane A; and if the ends of the plane had been reversed so that the deep end came forward, similar results would accrue, but as there would then be such an accumulation of pressure about the leading edge, it would be almost impossible to give a vessel with such a form a satisfactory sail-spread. With the sloping edge turned forward, a quantity of what may be termed perfect pressure, is graduated aft; this feature, coupled with the fact that the centre of gravity of the figure is relatively aft, instead of relatively forward, as it would be with the ends reversed, admits of a convenient and satisfactory arrangement of sail.

There is yet another strong argument in favour of a raking keel, which involves a question of speed. If the triangle *v* were taken away from A, the surface would be reduced one-fourth; and, consequently, the resistance to forward motion, dependent on surface friction, would be pro-

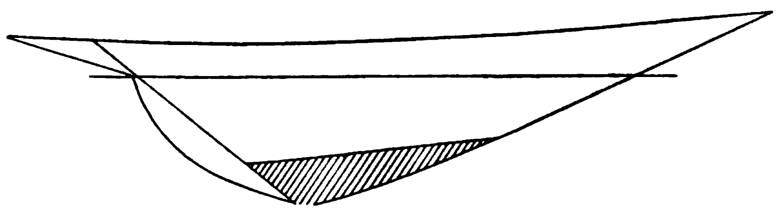


FIG. 51.

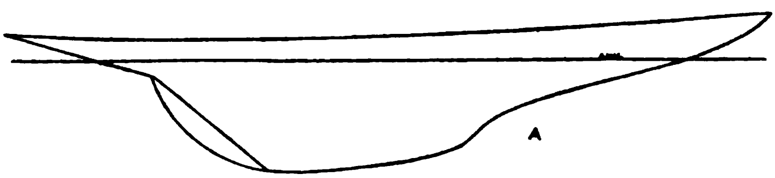


FIG. 52.

portionately reduced. It might not be prudent to make this reduction of surface if the effectiveness of the lateral resistance were going to be thereby reduced; but the fact is that the effectiveness of the lateral resistance would be almost unimpaired; a comparatively useless piece of wood would be removed, and a positive gain would ensue in the matter of frictional resistance. (A table for draught of water and area of lateral resistance will be found on page 19.)

The necessity of keeping the centre of lateral resistance, relatively to the length of the vessel, far aft, and the fashion of much raking the stern-post, led builders to by degrees advantageously increase the rake of keel, and there are many prevailing examples of disproportionate draught of water fore and aft. (Fig. 51.)

This form in turn gave way to that shown in Fig. 52, with a large hollow at A.

The piece at A was cut away to diminish the wetted surface consequent upon the greater depth at which the lead keel is now placed, and to the fact that the under side of the lead keel was made almost parallel to the load water line, in order that the general centre of gravity of the keel should be lower than it would in a form like Fig. 51.

One supposed disadvantage—especially in small boats—of a very much cut away fore-foot or of a triangular centre-plate instead of a deep fixed keel, is that in very disturbed water a vessel's head, in beating to windward, gets "knocked off the wind;" but it is overlooked that if the bow is readily knocked off the wind the same facility exists for "coming-to" the wind during favourable puffs.

A vessel with a much raked keel will probably steer wildly off the wind, and will require watching, and on any point of sailing she is likely to run off her helm. To meet these drawbacks, some eastern boats (such as those of Bombay) have cambered keels, *i.e.*, the reverse of rockered, as the back of the arch is turned upwards; and a few boats in America and in this country have been fitted with double centre boards. If, as before said, a modern vessel with raking keel is quick in falling off, she will be equally sensitive in coming to; and a careful helmsman will take her farther to windward in any given time than one of the older craft could be taken; and further, the helmsman will find the vessel with a raking keel, when sailing by the wind, a pleasant one to steer; she will answer her weather helm for a foul puff with wonderful quickness, or spring to quickly under a little lee helm for a free one. In stays they spin round like a top, and are off on the other tack almost before the head sheets can be handled.

Hitherto the question of leeway has been considered upon the assumption that a yacht has practically a level water surface on each side; but that is very far from the case, especially at considerable speeds. A yacht as she moves ahead creates a wave hollow on each side, and actually settles down into this hollow; and the surrounding water level may be, and usually is, above what would be the plane of flotation of the yacht if heeled whilst at anchor in smooth water. But the wave hollows are not uniform in size, and the larger may be either to leeward or to windward, according to the form of the yacht and the extent of her heeling. In shallow yachts the larger wave hollow is usually to leeward, and whether or not the yacht would settle to leeward as well as below the water surface would depend upon the size of the wave crest at bow and stern, and whether the increased pressure from them exceeded or not the decreased pressure caused by the wave hollow amidships. But in deep yachts of the British type the larger wave is usually to windward (see Figs. 53

and 54); the deep yacht will soak into the hollow (H, Fig. 54) which she makes to windward, providing, of course, that the pressure on the weather side is reduced below the pressure on the lee side. Whether the yacht actually settled to windward of her course would depend upon the lateral-horizontal component of the wind pressure on the sails (which produces leeway) exceeding the balance of the water pressures on the hull which tend to cause her to soak to windward into the wave hollow; but this tendency to soak into the wave hollow to windward must have the effect in some degree under any condition to reduce leeway if it does not extinguish it. No doubt these features help to account for the weatherliness of some narrow and deep yachts which are sailed at large angles of heel, and create a deep wave hollow under the weather bilge.

The effectiveness of triangular centre-boards on shallow vessels is well known, and it is astonishing how small a piece of board will check lee way,

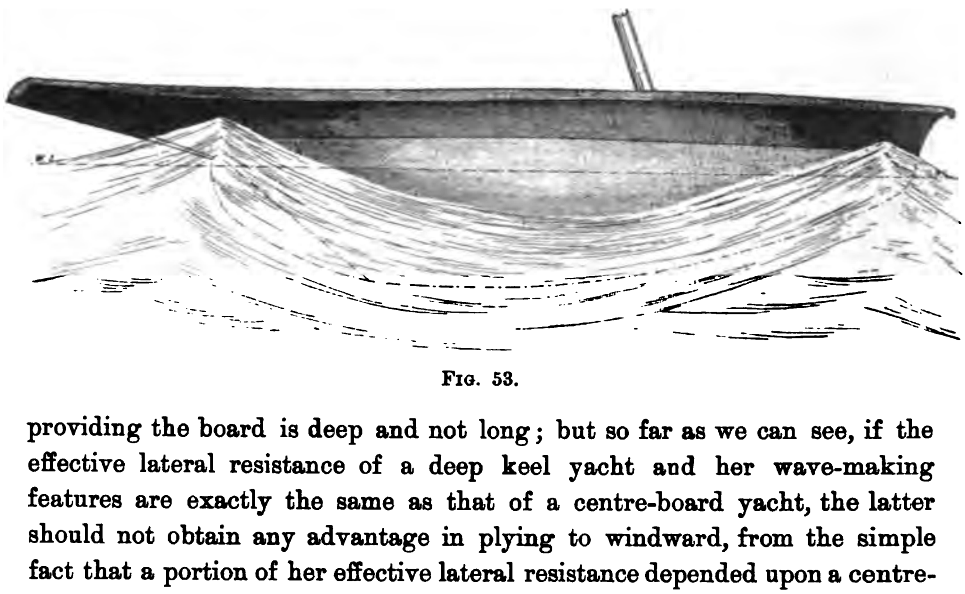


FIG. 53.

providing the board is deep and not long; but so far as we can see, if the effective lateral resistance of a deep keel yacht and her wave-making features are exactly the same as that of a centre-board yacht, the latter should not obtain any advantage in plying to windward, from the simple fact that a portion of her effective lateral resistance depended upon a centre-board. A centre-board, for instance, does not of itself enable a yacht to lay closer to the wind than a keel yacht can; but if two yachts of the same length and with the same sail spread are sailing by the wind, and one has a smaller displacement than the other, and carries her canvas better, she will travel faster through the water, or for the same speed be able to lay closer to the wind. The smaller displacement may be possible on account of greater breadth and smaller depth, the deficiency in draught of water being made up by a centre-board. Of course, if one vessel carries her canvas better than another, she will be more upright when

sailing, and her effective lateral resistance will be greater, whether she has or has not a centre-board.

Thus the effect of a centre-board will not be so apparent in holding a deep bodied craft with great rise of floor as the effect of a similar board on a yacht with a flat floor and shallow draught, for the principal reason that the centre-board in a shallow yacht bears a much larger proportion to the effective plane area for lateral resistance than it does in a deep bodied yacht with great rise of floor. Indeed, this proposition can be taken as a truism; but it does not mean a centre-board will have no effect at all on a yacht with great rise of floor, although we believe that was nearly the conclusion arrived at in the case of the Iverna. As a matter of fact, the influence of a centre-board in preventing leeway should be directly as its effective

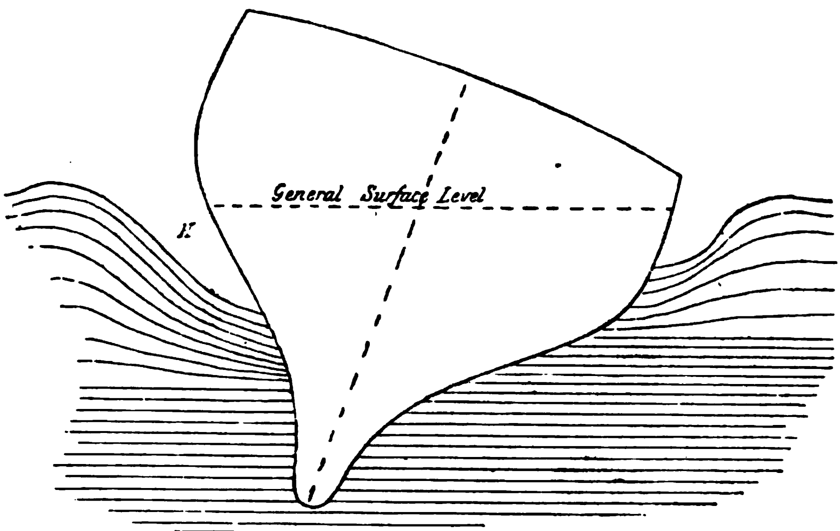


FIG. 54.

surface, or as its area and the angle it is employed at by reason of the heeling of the yacht. At a speed of eight knots an hour an English cutter of the deep-bodied type of 80ft. load line without a centre-board, has a lateral resistance which admits of an angle of leeway of about a quarter of a point, or nearly 3°, equal to 300 yards in a reach of three miles. If a board be added whose effective area is equal to one-sixth the effective plane area for lateral resistance of the yacht, the angle of leeway would be reduced one-sixth approximately, or, say, by 50 yards (making the angle of leeway 2½° only). In the case of the very shallow type of yacht, the area of the centre-board is often as much as one-third the area of the assumed plane of lateral resistance of the yacht (usually termed the "immersed vertical-longitudinal section" of the sheer plan); and we believe

that in the modern American type of yacht, like Volunteer, Mayflower, or Vigilant, the board area is about one-sixth the area of the assumed plane of lateral resistance. It would be almost impossible to accurately calculate the effective area for lateral resistance of a shallow-bodied yacht; but in a deep-bodied yacht with great rise of floor, it exceeds somewhat the assumed area.

From the foregoing it will be understood that the effect of a diminutive centre-board, bearing a relatively small proportion to the effective plane area (not to be confounded with the assumed plane) of lateral resistance of a deep-bodied yacht, has such a trifling effect on leeway that it might well escape observation. For instance, if leeway be only reduced to the extent of half a degree, the distance saved to windward on a reach of three miles would be about 50 yards, disregarding the retardation of speed due to the frictional resistance of the board and the water playing in the trunk. This latter retardation of speed, in a vessel with a very raking fixed keel, might be considerable, and more than counterbalance any advantage of slightly increased weatherliness due to the board. Thus, if six reaches were made, each three miles in length, the total distance saved would be about one-sixth of a mile, or a little more than a minute in time, if the speed equalled eight knots an hour. Of course, relative weatherliness in different yachts does not depend entirely upon resistance to leeway. The stability and wave-making features of the hull (see pages 72 and 76), and frictional qualities of the skin, the sit and trim of the sails, and the capabilities of the helmsman would have a great deal to do with the matter; but if all these things were equal, then the saving in time, by adding a centre-board as described to a deep-bodied yacht, would be as we have stated, disregarding any retardation of speed due to the frictional resistance of the board and the obstruction offered by the trunk.

However, for reasons not easily explainable, the qualities are often not equal, even in cases where there was every reason to suppose they would be. We frequently find a yacht of superior under-water depth and area of surface for lateral resistance, not so weatherly as another of inferior depth and area as stated. Also it frequently happens that of two yachts of similar length, breadth, and draught of water, that the one of lighter displacement will be the more weatherly; or it may be that the heavier of the two shows this quality in a greater degree. In such cases it will generally be found, that if the yacht of lighter displacement is the more weatherly, that she is the stiffer of the two; she either carries a greater weight of ballast in proportion to the total displacement, or has a higher centre of buoyancy, or both. On the other hand, the yacht of heavier displacement might excel in those attributes, and in such a case ought to be more weatherly.

But the cause of a deficiency in weatherliness may be attributable to some fault in the form of the hull of the yacht, or to the make, disposition, or sit of her sails. As already stated, if a yacht makes a large hollow wave to leeward along her mid-length, she must lose somewhat in general pressure on that side, if the loss should not be balanced by an increased pressure from the wave crests at bow and stern (see page 76), and her lee way would be thereby increased; but if the large hollow wave is made to windward, that might be a cause for increased weatherly qualities, as the

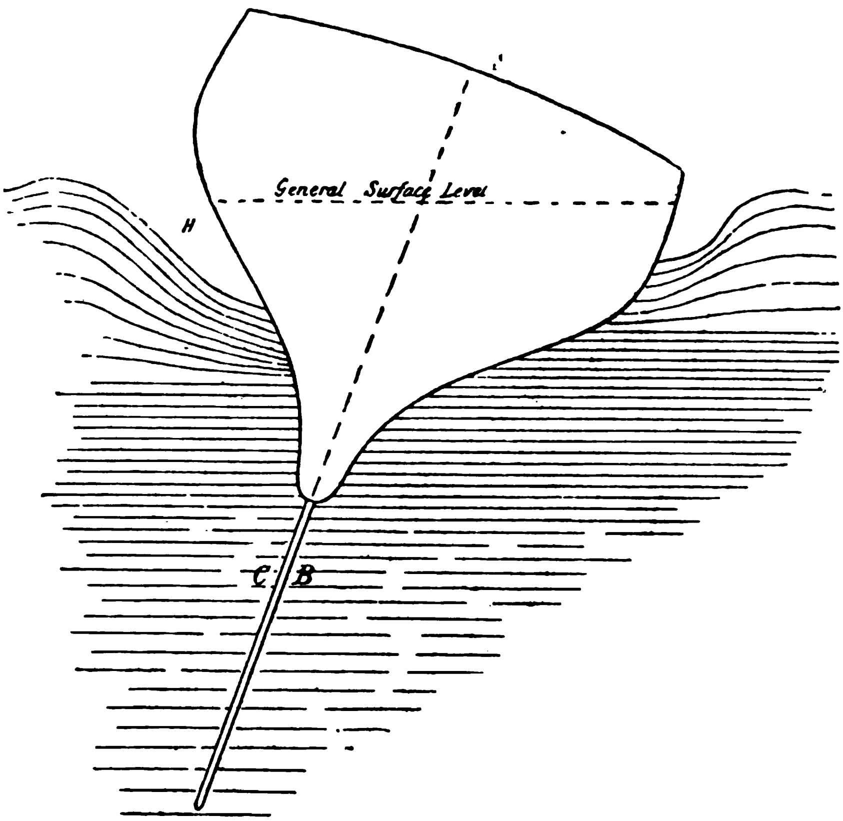


FIG 55.

removal of pressure on the weather side would be equal to an increase of pressure on the lee side, and cause the yacht to soak to windward into her wave hollow. (See Fig. 54). These features would, however, be subject to some modification in a yacht fitted with a centre-board. In the case of a shallow yacht making the larger wave to leeward, a board deeply immersed in solid water would, no doubt, to some extent prevent her sagging into it; and a similar effect of the board (see Fig. 55) will prevent a deep yacht of large displacement soaking into the wave hollow

which she makes to windward. This, no doubt, will in a considerable degree explain why, in a deep heavy yacht with comparatively large wave-making features, the board is apparently less effective than it is in a lighter and shallower yacht with smaller wave-making features; in other words, the board might make a heavy deep yacht less weatherly than she would be without it, irrespective of the obstruction it would offer to speed.

With regard to the relative merits of keel and centre-board yachts, it is contended, with some truth, that a small keel yacht, say of 40ft. load line, will appear to more advantage against a centre-board yacht of equal

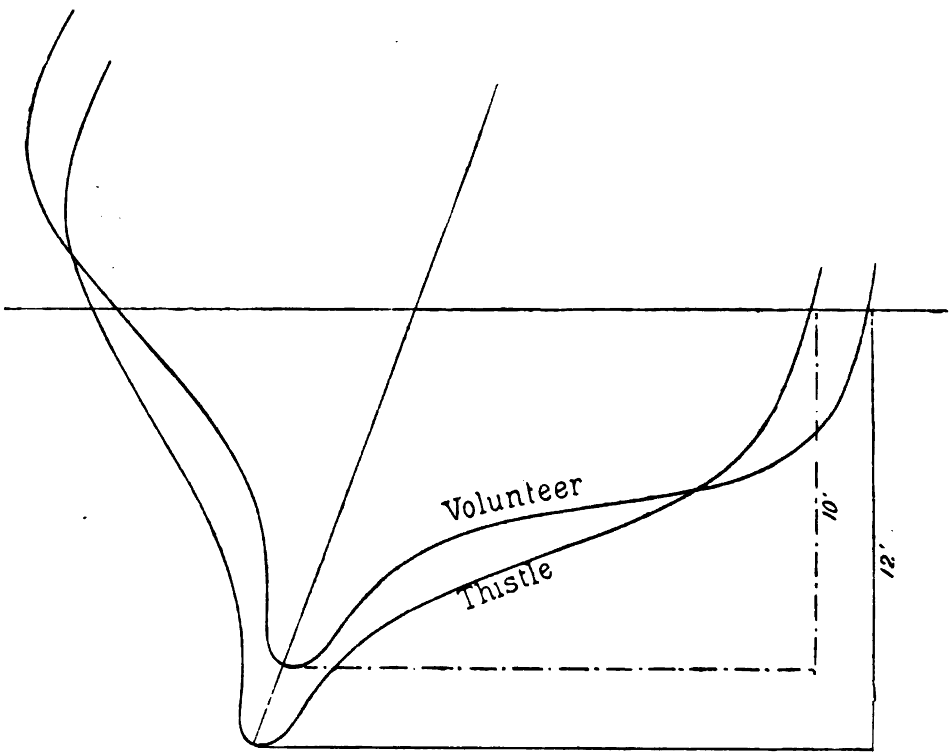


FIG. 56.

length, than a keel yacht of 80ft. load line does against a centre-board yacht of 80ft. load line, for the reason that the small keel yacht has greater relative under-water depth. This was particularly noticeable in yachts built to suit the old tonnage rule, owing to the necessity of great under-water depth to compensate for narrowness of beam; and although a narrow yacht could be given a depth which would admit of a range of stability much longer than that of the shallow centre-board yacht, she could not be made equal to the centre-boards in the matter of initial stability, a quality which under some conditions largely governs weatherliness. However, under the

existing rules for the rating of yachts, there is no restriction on beam, and the fixed keel yacht can be made as stiff (or stiffer) at initial angles of heel as the broader centre-board yacht.

Mr. B. Martell made a comparison of the effective lateral resistance of Thistle and Volunteer, and he computed that Thistle's effective surface at 16½° is 780 sq. ft., and that of Volunteer at 15° (minus her centre board), 634 sq. ft., or, adding the 134 sq. ft. (area of the centre-board), 768 sq. ft. At 20° inclination their relative effective surfaces are as 10 : 12 (see Fig. 56), and with Volunteer's centre-board down they are about equal; also, the immersed half girths (to leeward) of the two yachts are equal (20ft.), and so it can be assumed the frictional resistance to leeway would be alike in both. It thus seems plain that we must look beyond the calculated effective surface of lateral resistance of these two yachts for an explanation of the superior weatherly qualities displayed by Volunteer. If the superiority is to be found in the hull, it is probably traceable to the fact that the wave-hollows on the lee and weather side of Volunteer were about equal, and not so deep as those in Thistle; and if there were any inequalities in the pressures tending to allow the yacht to slip into the hollow to leeward, her centre-board acting below in undisturbed water would prevent it; and to this might be added that, in all probability, at 20° inclination, the general head resistance of Volunteer would be the smaller, although it seems certain that in the upright position Thistle showed the greater speed.

Another problem connected with the sailing of large centre-board yachts in tideway channels with restricted depth of water, is one dependent on the varying velocity of the flow of water on the surface, and at depths below the surface. In 1856 some experiments were made on the Thames, to ascertain the change in velocity due to depth, the experiments being carried out on the measured mile in Long Reach, both on the flood and ebb. As the ebb runs with the greater velocity, and is the stream with which racing yachts will be mostly concerned, we will only deal with that. It should be mentioned that the average depth of the channel in Long Reach is about the same as the average depth in Sea Reach, viz., about 30ft. at low water, and 45ft. (for an average tide) at high water. The experiments showed that the ebb 1ft. below the surface had a velocity of 2·94 knots an hour; and at 10ft. below the surface 2·72 knots an hour; and at 20ft. below the surface 2·58 knots an hour. The tide at the surface, therefore, ran nearly half a knot an hour faster than it did 20ft. below the surface; and at 10ft. below the surface about one-seventh of a knot an hour faster. Now, if a centre-board yacht with 20ft. draught of water be beating to windward with a weather-going tide down the Thames, she would only be

aided by the mean velocity of the stream, equal to 2·75 knots instead of 3 knots; if she lifted her board and reduced her draught to 10ft. or 12ft., the mean velocity of the current she would be favoured with would be 2·84 knots or one-tenth of a knot greater. The question here would be whether the reduction in leeway, due to the influence of the board, would equal the loss due to its immersion in the slower current. Of course, with a lee-going tide the case would be exactly reversed, and the deeper the board could be immersed, the greater its influence would be in checking the effect of the faster current on the surface. These features are well understood among sailors, or at least it is a matter of common belief amongst them that a craft of deep draught does relatively better to windward against one of shallower draught when working with a lee-going tide. It will be gathered from the foregoing that the true effect of a centre-board on a vessel's normal leeway might easily be obscured by causes due to the variation of the velocities at different depths of the stream in which she happened to be working. The true effect of the board could, however, be nearly ascertained at the time of slack water. The general conclusion is that the board should not be made full use of with a weather-going tide; and that it should be dropped to its last inch with a lee-going tide. A secondary effect of the board in the case of working with a lee-going tide will be produced from this cause. With a lee-going tide a vessel is continually slipping away from the wind or receding from it; whereas, with a weather-going tide, she is pressed against the wind. The effect of this is very considerable in plying to windward. (This matter will be found fully explained in the next chapter). In the Solent, or West Channel, the depth of water is pretty much the same as it is on the Thames, and in all probability the current velocity varies much about the same at different depths; but most likely there would be a different variation in the velocities of flow in deeper channels.

CHAPTER VI.

POWER TO CARRY SAIL—THE IMPULSE OF THE WIND AS A PROPELLING POWER.

The power of a vessel to carry sail is dependent on her statical stability; and, consequently, if the stability is known, so also can the area of sail which a vessel will carry with a given wind pressure be determined. Thus, to heel a vessel to any given angle, the moment of the sails must be made to equal the moment of stability at that angle of heel. For illustration, the sails can be regarded as a plane surface, and the wind as a force acting at right angles to that surface; therefore, it is assumed that the sails are trimmed exactly fore and aft. The resultant force of the wind acts through the collective centre of gravity of the sails, or through the centre of effort of the sails, as it is termed, on a coupling lever, the length

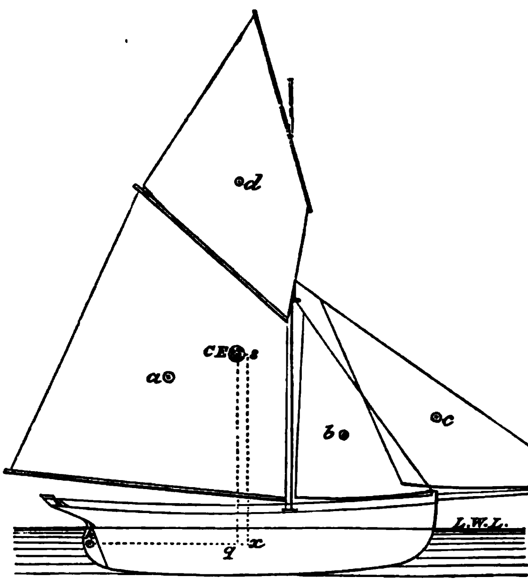


Fig. 57.

of which is the height that the centre of effort is above the centre of lateral pressure of the water of the hull, termed the centre of lateral resistance.

If the wind always blew at right angles to the plane of a sail, the effort of that wind would be exerted through the centre of gravity of the sail plane; that is, if the whole effort of the wind were concentrated on one point of the sail, that point would necessarily be the centre of the plane. This point is usually termed the "centre of effort" of a sail. If a vessel has many sails, such as a cutter yacht, Fig. 57, the total effort of the sails is exerted collectively through a point which represents the centre of gravity of the *whole* of the sails combined. In Fig. 57 the centre of effort of each sail is marked at *a b c d*, and the common centre of gravity of the four sails lies at the point C E, and this point is termed the CENTRE OF EFFORT of the sails.

In computing the moment of the sails, the following notation will be observed:

A = Area of sail.
B = Height of centre of effort of sails above centre of lateral pressure on the immersed portion of the vessel.
C = Pressure of wind per lb. per square foot of canvas.
x = Moment of sails.

Then $$x = A \times B \times C.$$

Mr. Fincham calculated that in yachts 1lb. pressure per square foot of lower sail ought not to cause more than an inclination of from 6° to 9° in yachts, but the improvement in ballasting and increase of depth make any rule for providing for safety quite unnecessary. Thus, with a wind strength equal to 1lb. pressure per square foot, a modern cutter of 155 tons displacement would carry gaff topsail and jib topsail, and her whole area of sail would be about 8300 square feet. The centre of effort of the sails would be 55ft. above the centre of lateral resistance; then

$$\begin{array}{r} 8300 \times 55 \\ 55 \\ \hline 41500 \\ 41500 \\ \hline 456500 \end{array} = \text{moment of sails.}$$

This sum, divided by the displacement (expressed in pounds, or 155 tons × 2240 = 347,200lb.) will give the length of the righting lever the cutter should have to balance the area of sail just mentioned with 1lb. per square foot wind pressure on it.

$$\text{Righting lever} = \frac{456500}{347200} = 1{\cdot}32\text{ft.}$$

This is nearly the length of the righting lever which the cutter with 3ft. metacentric height would have for an inclination of 26°; and 26° is about

the angle of heel the cutter would take with such canvas on her as described; and if it carried her to 45° she still would be in no danger of capsizing.

With lower sail only, the sail moment would be as under (taking the pressure as 2lb. per square foot). The area of her lower sails would be about 5600 square feet, and their centre of effort 44ft. above the centre of lateral resistance; then $5600 \times 44 \times 2 = 492{,}800$ = sail moment, which, divided by the displacement in pounds, will give the length of the righting lever—

$$\text{Righting lever} = \frac{492800}{347200} = 1{\cdot}33\text{ft.}$$

which is about the length of righting lever at 26° for a metacentric length of 3ft.

Several rules have been laid down as ready measures of a vessel's sail-carrying power; but as actual, or even approximate, stability is not a quantity in any of the rules, they are of little value.

The only trustworthy formula for determining beforehand the area of sail suitable to any given vessel is that which includes the height of the metacentre above the centre of gravity. It has been shown that, if a vessel's metacentric height is known, her stability can be closely approximated for any angle of heel it will be desirable to sail her to in a given wind pressure. The righting moment for any given angle of heel will be

$$\text{Righting moment} = D \times M\,F \times \sin.\ \theta.$$

D is the displacement in pounds avoirdupois; M F height of metacentre above the centre of gravity; sin. θ, sine of angle of heel.

The cutter's moment to carry sail at 26° inclination will be thus computed:

```
    155  tons displacement.
   2240  lb. (= 1 ton.)
   ----
   6200
   310
  310
 ------
 347200  × M F (= 3ft.)
      3
-------
1041600  × sin. θ (= ·438).
   ·438
-----------
   8332·800
  31248·00
 416640 0
-----------
 456220·800 = righting moment in foot pounds.
```

This moment, divided by an assumed wind pressure per square foot of sail and a given height for C.E. above the C.L.R., will determine the sail area which could be carried at the given angle of heel. (*See* page 81.)

Hitherto the effective sail area at any given angle of heel under any given wind pressure has been regarded as constant; and so, indeed, it is

nearly up to 10° or so; but, as a matter of fact, the effective sail area is constantly decreasing in the ratio of the square of the cosine of the angle of heel; that is, the area is decreased in proportion to the cosine and the length of the heeling lever (height of C.E. above C.L.R.) in the same proportion; therefore the heeling force becomes reduced as the square of the cosine of the angle of heel. The effect of this reduction can be shown by a curve. It is assumed that the schooner Lyra's lower sail area of 7100 square feet is acted upon by a wind pressure of 2lb. per square foot; her C.E. is 46·5ft. above C.L.R. This gives her sail moment as 300 foot-tons for the upright position, and Lyra would heel until this moment became balanced by the moment of stability. It will be seen that this ought to occur at 13°, but, in reality, she would not be taken to that angle, but to the angle where the two curves intersect at 12° 15′. (*See* dotted line, Fig. 58.)

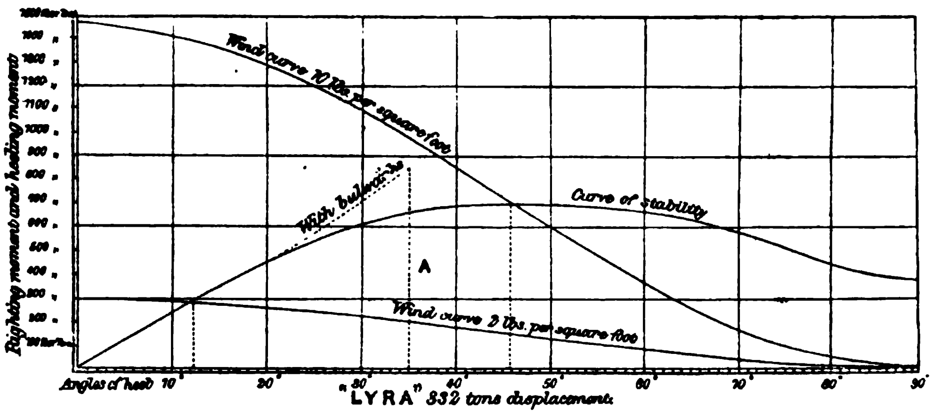


FIG. 58.

The effect of a very much greater pressure of wind, equal to 10lb. per square foot—the pressure of a heavy gale—is shown by the second wind curve, which assumes the yacht to be "caught" abeam, with four lower sails set. Such a pressure would probably knock the yacht down flat on her beam ends if applied suddenly, and if her sails did not split or rigging and masts give way; she would then, however, recover herself, and stick at an inclination of 46°, as pointed to by the dotted line, until the wind pressure moderated.

The bulwarks, it should be pointed out, will check the heeling to sudden squalls, as will be seen by the diagram. The dotted line A shows the angle where the bulwarks become immersed, and the increased resistance to heeling they would afford. Bulwarks, however, must not be relied upon as an addition to stability, as they might prove treacherous. The instant the rail is under water the yacht would be, in effect, reduced to her actual free-

board, and would take a sudden increase of heel or lurch which might be highly dangerous in some vessels.

From the foregoing will be gathered the value of stiffness in a racing yacht, as by standing up to her canvas she will have an undiminished sail area, and consequently undiminished propulsive power.

The effect of the diminished heeling force of the wind consequent upon the angle the sails may be trimmed with the wind has not been considered.

In sailing before the wind very diminished speed is attained to what may be realised with the wind abeam, assuming, of course, the sail area to be the same; the reason is that the velocity of the wind, or its effective force, is diminished by exactly the speed of the yacht; or say the real wind is travelling at the rate of 14 miles an hour, and the yacht at the rate of eight miles an hour, then the *apparent* wind, which is impelling the yacht, will only have the speed of 6 knots past her. The effect of the *apparent* wind on a yacht can be thus briefly explained: a yacht with 1000 square feet of immersed surface would meet with a gross resistance in the water of about 400lb. at a speed of 8 knots, and would have about 2000 square feet aroa of canvas. If the wind moved at a velocity of 14 knots, it would exert a pressure on a *fixed* sail of about 1lb. per square foot; but, as the sail is not fixed, but moves away from the wind, the pressure is diminished until it balances a resistance met with by the yacht in the water at a certain speed, say 8 knots. At this speed the resistance of a yacht would probably be increasing as the cube of the speed; and as the pressure on the sail would be diminishing as the square of the speed, it is quite evident that a limit to an increase of speed before the wind is very suddenly reached. Thus, say it were desired to increase the speed of the yacht to 10 knots, the resistance she would meet at that speed would be about 800lb., and as the wind pressure on the sail would be only that due to a velocity of 4 knots ($14 - 10 = 4$), the sail area would have to be increased to about 8000 square feet to maintain the 10-knot speed of the yacht. Beyond this we can imagine that the sail area could be increased until the yacht attained a velocity equal to the velocity of the wind, when no further increase of speed could be attained.

It can be seen that the reason why a vessel is driven ahead by the action or pressure of wind on the sails would need no explanation, if her course were always such that the wind blew over her stern. But the fact is, a vessel can be made sail within an angle of 45° of the direction of the wind. Thus, if a vessel had to arrive at some point in the line from which the wind was blowing, she could do so by traversing a distance that exceeded the distance from the starting point to the point to be reached as

1 is to 1·414 or $\frac{5}{12}$ increase of distance. In the annexed diagram (Fig. 59), let W be the direction of the wind, A the point of departure, and B the point to be reached, 14 miles distant. A C makes an angle of 45° with A B, and a vessel could proceed along the line A C. By tacking or altering her direction at C, she could sail along the line C B, and so arrive at the point B, her destination. Then the distance traversed will be of course

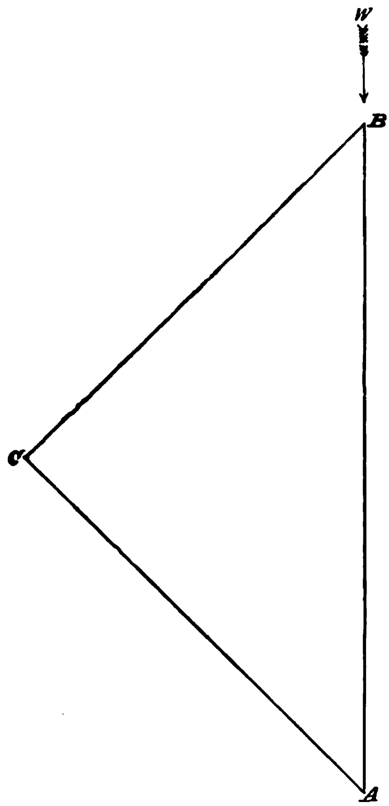


FIG 59.

equal to A C and C B; or two sides of the triangle A B C. The length of the side A C can thus be solved: A C $= \frac{\text{A B sine B}}{\text{Sine C}}$; or, as the two sides A C, C B are equal, the distance traversed will be $\frac{\text{2 A B sine B}}{\text{Sine C}}$, giving the quantities.

·707	A B............... =14 miles.	
14	Angle A .... =45°	sin. = ·707.
2·828	Angle B .... =45°	sin. = ·707.
7·07	Angle C .... =90°	sin. = 1·000.
1)9·898		
2		

19·796 = distance traversed to reach the end of the line A B, Fig. 59.

If a vessel lie $3\frac{1}{2}$ points from the wind, she will traverse 18 miles; and if she lie $4\frac{1}{2}$ points from the wind, 22 miles. Therefore, in beating *dead* to windward, the proportion the real distance to be reached bears to the distance traversed will be as 1 : 1·3 for $3\frac{1}{2}$ points; 1 : 1·414 for 4 points; and 1 : 1·6 for $4\frac{1}{2}$ points. Thus, if a vessel laid $4\frac{1}{2}$ points from the wind, and had to beat 20 miles dead to windward, she would sail a distance—

5 points	20 × 1·826 = 36·5 miles.
$4\frac{1}{2}$ points	20 × 1·6 = 32 miles.
4 points	20 × 1·414 = 28·3 miles.
3 points	20 × 2·3 = 26 miles.

This does not take into account leeway or currents.

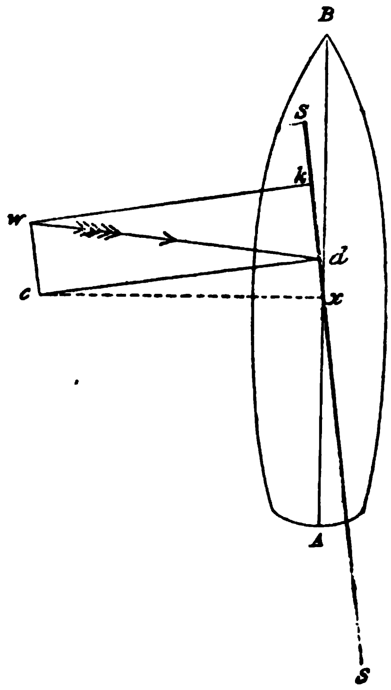


FIG. 60.

Now, the cause of the vessel proceeding along the line A C, instead of with the wind, in the direction W B A, or dead to leeward, can be explained.

In Fig. 60 we will suppose A B to be a motionless boat with a balance lug sail set. The line *w d* represents on a scale the direction and force or speed of the wind equal to 10 miles an hour, known as a moderate breeze. The line *s s* represents the balance lug sail of an ice boat. It is obvious that the wind, blowing on the sail from the direction shown, would tend to drive the boat in the direction of its own motion. But the force *w d* is

made up of two components—one, *c d*, is acting at right angles to the sail, and the other, *d k*, acting along the plane of the sail without any potent effect. The component *c d* can be further regarded as a force made up of three other components—one, *c x*, acting at right angles to the keel of the vessel, and tending to drive her to leeward; another, *x d*, acts in a line with the vessel's keel, and tending to drive her ahead. The third force is resolved vertically in a downward direction, and need not be considered.

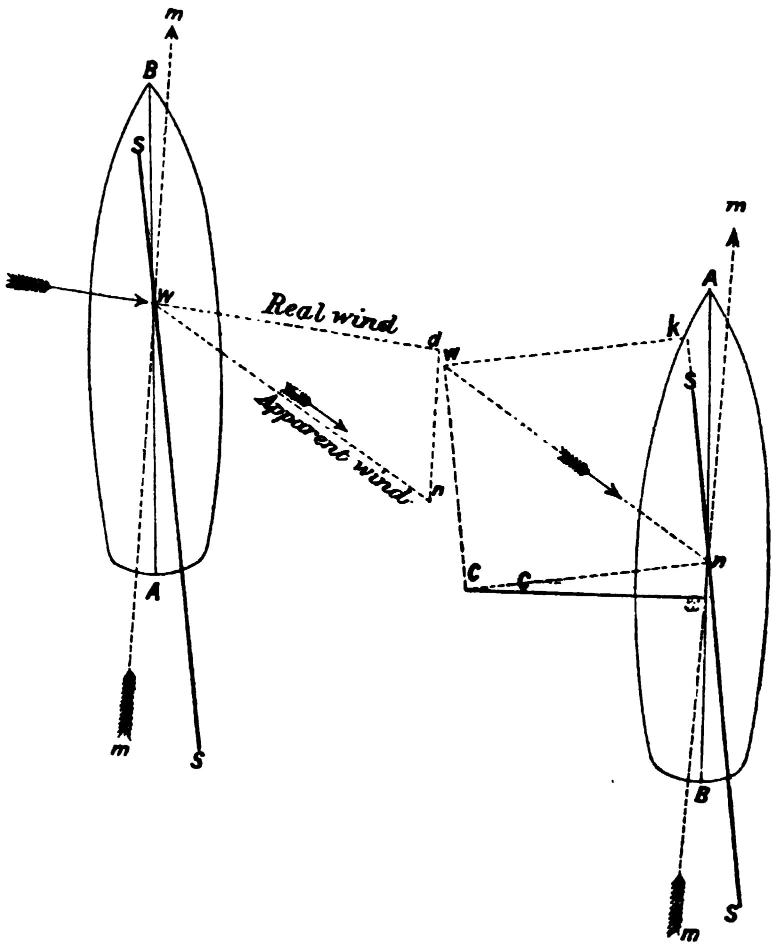


FIG. 61. FIG. 62.

We have now to consider what takes place when a boat, under the influence of *x d*, gathers way, or commences to move in a direction to meet the wind at a more or less acute angle. In Fig. 61 A B is the boat set in motion by the component of the wind *x d*, and is proceeding in the direction of the arrow *m m*; the difference between the arrow *m m* and the keel line represents the angle of leeway, or amount of leeway made. The dotted

line *w d* represents the direction and velocity of the real wind, as in Fig. 60; but the wind *apparently* will not be blowing more ahead as the boat gathers speed, and it is the *apparent wind* with which we have to deal. The apparent motion of the wind is the resultant of the real motion, and of a motion equal in force and opposite to the motion and speed of the vessel, and is thus determined: On a line parallel to *m m* set off a distance (*see d n*, Fig. 61) by scale (same as the wind scale) to represent an opposing motion *equal* to the speed of the boat (assumed in this case to be five miles an hour, or half the velocity of the wind). Join *n w*, and the dotted line *n w* will represent the force and direction of the *apparent wind*.

This apparent wind must now be regarded as the propelling force, and not the real wind as shown in Fig. 60. In Fig. 62 let the dotted line *w n* represent the direction and force of the *apparent* wind; by a parallelogram of forces the line of force *w n* has two components, one acting in the direction *w c*, or *k n*, and the other in the direction *c n*. The component *c n* is farther resolved into three components, as before shown by Fig. 60, represented by *c x* and *n x* in Fig. 62, and another acting vertically, not shown. It is the component *n x* which impels the boat forward. It will be seen that the force *n x* is very small, or only about one-seventh of the force *c x*, which is striving to drive the boat to leeward; but the resistance to leeway is very great, whilst the resistance to headway is very small. Consequently the boat moves a scarcely perceptible distance in a broadside direction, but gathers speed in the direction of her keel, or rather in the direction of the line *m m*, which includes the broadside motion or leeway. The boat continues to gather way or increase in speed from the rest position shown in Fig. 60 until the resistance she meets with, from friction, wave-making, and air-resistance, &c., equals the force shown by *n x* (Fig. 62). The speed of the boat then remains uniform so long as the wind is constant.

If a vessel resembled a true hemisphere, with no keel or dead wood of any kind, it is certain that she would, under the influence of a wind pressure, proceed to leeward in the *c n* direction much faster than ahead in the direction *m m*; but a yacht is so formed that she offers very great resistance to lateral or sideway motion, and a very little to headway. In similarly formed vessels, the difference between the resistance to lateral motion and to forward motion is generally taken as proportional to the area of the midship section and the area of the longitudinal section, it being always understood that only the immersed portions of these sections are referred to. This proportion roughly is as 10 to 1; but as a fact it bears very little relation to the actual value of the lateral resistance and resistance to forward motion. The latter in reality depends upon the form of the vessel and her area of immersed surface. For speed of 6 knots this

relative value will be better found by simply comparing the resistance due to surface friction and the resistance to broadside action. Thus, say a 40-rater has 1100 square feet area of immersed surface, and a plane of 470 square feet area for lateral resistance; at six knots the frictional resistance on the immersed surface, consequent on forward motion for moderately smooth copper, will be equal (nearly) to 0·25lb. per square foot, or 275lb. in the aggregate. That is, the whole force required to move the vessel at a speed of six knots will be 275lb., that being the total resistance at such a speed.

The resistance on the plane of 470 square feet to *lateral* motion will not be one of "friction," but of direct pressure, and if a yacht could be moved sideways at a speed of six knots the resistance would be 112lb. per square foot, or 52,640lb. in the aggregate. Also in sailing at a broader angle than 45° $n\ x$ would become a larger fraction of $c\ n$. In the diagram (Fig. 62) let $n\ x$ represent a force of 275lb. which overcomes the resistance to forward motion, and $c\ n$ a force which is three times 275lb. or 825lb. The force of 825lb. is exerted on a plane of 470 square feet area, which is equal to 1·7lb. to the square foot, and the resistance that would balance a force of 1·7lb. per square foot would be met with at a speed of 0·78 knot. Then, if the headway were six knots, and the leeway 0·78 knot, the "angle of leeway" would be 7°. But this would be too much, and an allowance would have to be made for the loss of sideway pressure due to the forward velocity of the yacht and her wave-making features, the broadside motion varying in fact with the speed of the yacht. We are not aware if anyone has actually tested the leeway a yacht will make when close-hauled under a wind-pressure that will give her headway at the rate of six knots an hour; but it has been variously estimated from 3° to 11°, or from ¼ point to 1 point of the compass. We are inclined to think, so far as our observation goes, that a cutter yacht of, say 80ft. water line, when sailing a course 4 points from the wind, will, including leeway, make a course of about 4¼ points, if her speed be 6 knots through the water; it would, however, be impossible to ascertain this accurately, as the wind does not remain sufficiently constant in strength and direction even whilst a mile could be traversed. Small yachts relatively make greater leeway than larger vessels, but the difference is not so great as might be expected.

It will be gathered from what has already been stated with regard to the relation of the *apparent wind* to the real wind, that the effective angle for the sails to the line of motion of the vessel must largely depend on the speed. This is well illustrated by the case of ice yachts, which at times have been known to exceed the velocity of the real wind for miles together; in fact, there are authenticated cases of an ice yacht having sailed 9¾ miles

in 10 minutes* with an actual velocity of the real wind of 30 miles an hour; that is to say, the speed of the yacht was nearly double that of the wind. Upon reference to Fig. 61 (page 90), it will be seen that if *d n*, which represents the speed of a sailing vessel, were made longer, the line *w n* representing the *apparent wind* would become longer also, and make a more acute angle with the line of motion of the vessel, and the sail would require to be trimmed more in a line with the keel. In the case of ice yachts the greatest speed is made when the real wind is blowing on the quarter or considerably abaft the beam, and the highest speed is attained when the *apparent wind* appears to be nearly right ahead. It is obvious, under such conditions, even allowing for the augmented force of the *apparent wind*, that the effective force must be very small, and were it not for the circumstance that an ice yacht meets with an infinitesimally small resistance to headway (a touch of the finger will set them in motion when at rest), their phenomenal speed would be impracticable, as it is in the case of yachts which have to sail with a portion of their body submerged in water. The two cases can be illustrated in this way: the only resistance an ice yacht meets with is from the friction of the runner on the ice, and when sailing at a high speed only the lee runner comes in contact with the ice owing to the weather one being lifted by the heeling of the yacht; this friction, too, is of a very fine quality, and nothing resembling the frictional resistance of water. The sailing vessel exposes to the frictional resistance of the water the whole of her immersed surface, which in a yacht of 20 rating will be nearly 700 square feet. The magnitude of this resistance will be better understood if the immersed surface is assumed to be a plane, spread out flat on the face of a perfectly smooth sea.

However, although it is not necessary in the case of sailing vessels to consider such cases as sailing faster than the wind, yet so much has the resistance of yachts been reduced by the reduction of immersed surface, and so enormous has their sail area become, that they sail very much "nearer the wind" than they formerly did; this simply means that they make a greater speed in relation to the real wind; and "close-hauled" in one yacht may mean 2½ points off the direction of the *apparent wind*, or as much as 4½.

In the table on the next page the angle for the sails of a yacht with the keel is given in column 6 for any given angle the yacht may make with the real or *apparent wind*, as set forth in columns 3 to 5. This table has been calculated from a formula given by the late Professor Rankine,†

* See "Yacht and Boat Sailing," in which the whole phenomenon of the speed of ice yachts is explained.

† See his "Shipbuilding, Theoretical and Practical."

who assumed that the normal pressure of the wind varies as the square of the sine of the angle of incidence (which strictly it does not, although the assumption is correct enough up to the angles sails are trimmed for close-hauled sailing). He pointed out that sails were most advantageously trimmed when the tangent of the angle they make with the *apparent* wind is double the tangent of the angle they make with the vessel's course. He further pointed out that this angle for the sail can be determined thus: In Fig. 63 the base line N O will represent the vessel's course; then, with that line for a diameter, describe the semicircle N P O; then make a smaller semicircle with a diameter Q O, one-third the length of N O. From Q draw Q *c*, representing the *direction* from which the *apparent* wind blows; then from O produce O *c*; join Q *d*, and Q *d* will be the required angle for the sail. (See column 6 in the table below.) The angle made by the foot of the sail is referred to, but often the angle

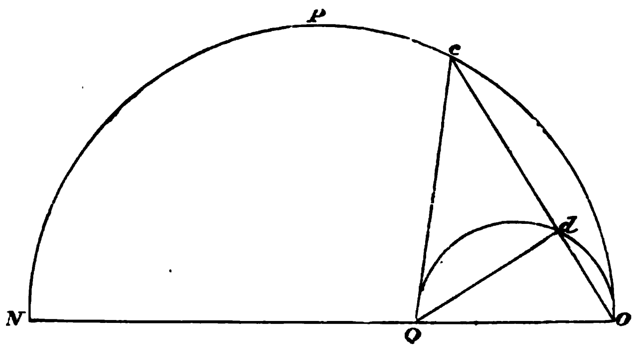


Fig. 63.

made by the gaff may be greater; and if the gaff goes off very badly the boom will have to be hauled in proportionally closer, so as to keep the sail at an effective mean angle. This is a matter well understood by those who have experience in fore and aft sailing.

True Speed of Yachts in knots.	Angle of course of Yacht to *real* wind in compass points.	Angle of course of Yacht to *real* wind in degrees.	Angle of course of Yacht to *apparent* wind in compass points.	Angle of course of Yacht to *apparent* wind in degrees.	Angle for sails, with the keel.	Velocity of *real* wind in knots.	Velocity of *apparent* wind in knots.
6·50	3½	39° 22′	2¼	28°	9° 20′	16	21·5
7·30	4	45°	2¾	30° 50′	10° 15′	16	21·8
8·00	4½	50° 37′	3	33° 30′	11° 30′	16	22·0
8·75	5	56° 15′	3¼	37°	12° 30′	16	22·1
10·00	6	67° 30′	3¾	42° 20′	14° 40′	16	21·8
11·20	8	90°	5	55°	19° 30′	16	19·6
11·40	10	112° 30′	6¼	72°	27°	16	15·6
10·40	12	135°	8½	94°	37°	16	11·3
8·90	16	180°	16	180°	90°	16	7·1
1	2	3	4	5	6	7	8

The velocity of the *apparent* wind set down in column 8 will help explain, in connection with what has gone before (*vide* p. 94), why there should not be greater difference in the speed obtainable from close-hauled sailing and with the boom broad off the quarter. The force of the *apparent* wind increases very rapidly as the vessel's speed bears a nearer proportion to the speed of the real wind, and the sails become trimmed flat in; and if a vessel only had enough canvas to propel her at a speed, say, one-third the velocity of the wind, her greatest speed would be with the wind on the quarter, or when twelve points from the wind, as in the case of ice yachts previously referred to.*

In the table given the greatest speed is shown when the yacht has the *real* wind two points abaft the beam, the *apparent* wind showing a little forward of the beam. This no doubt is according to practice, as, and it should be noted that, when the *apparent* wind is, judged by the vane on board a vessel, abeam, the real wind is always a little abaft the beam; in fact, the vane shows the direction of the *apparent* wind. The American yacht Sappho is said to have made sixteen knots for several consecutive hours, in crossing from New York to Queenstown, the wind at the time blowing with a velocity of from twenty-four to thirty knots in the hour. Her log shows the *apparent* direction of the wind, judged by the dog-vane, to have been S. and S. by W., and her course E. ¼S.; this would place the *real* wind two or three points abaft the beam, and general experience points to the conclusion that a vessel makes the highest speed with the real wind three or four points abaft the beam; and this will be more patent if the vessel be deficient in stability, as she will keep more upright under such conditions.

Of course, if the vessel be ill adapted for high speed, she may relatively make greater progress with the real wind well forward of the beam; this will be noticeable in some full-bowed vessels with a good hold of the water and fine after-bodies; at the slower velocities, on a wind, they will relatively do better for fast vessels than when reaching off the wind. The reason is, that they nearly attain their maximum speed on a wind; and the additional exertion of the wind, when its impulse is more direct to the plane of the sails, has but small effect in overcoming the increased resistance the vessel meets with in the water at higher speeds, on account of the fulness of the vessel's entrance.

The effect of sailing with or against the tide or current is frequently discussed, and it is often thought that the tide accelerates exactly as it retards a vessel's motion over the ground; but this is by no means a true

* The angle that a yacht makes with the real wind can be readily read off in tacking by taking half the number of points her head sweeps through in going from one tack to the other.

state of the case in either sailing on or off the wind. If, say, a yacht is sailing with the wind forward of the beam at the rate of 5 knots an hour, and suddenly has a current of tide with her equal to 3 knots an hour, her speed over the ground will have an increase due to the tide of exactly 3 knots, and she will meet the wind with all the greater force due to the accelerated speed. Upon reference to Fig. 61, page 90, it will be seen that *d n* can represent the speed of the yacht at 5 knots; and, if that speed be increased to 8 knots, the line *w n*, representing the *apparent* wind, will necessarily be increased in proportion, and draw more ahead, or make a sharper angle with the vessel's course. The immediate effect of this would be that, as the *apparent* wind came more ahead, the sails would have to be trimmed in flatter, and the yacht would make a course nearer the real wind, with but little or any loss of speed through the water; or, in other words, her "speed to windward" would be increased. A more general result, however, would be this: the helmsman, seeing the sails began to lift, owing to the *apparent* wind drawing more ahead, would, from habit, sail her harder, until her sails were full, and there would then be a small actual gain in speed through the water, which would more than compensate for the loss due to sailing broader off the *real* wind; and this compensation would exist without estimating the actual gain made to windward by the set of the tide. Inversely, if the yacht had sailed into a 3-knot current opposed to the direction of her motion, her speed would be retarded, as there would be a loss of propelling force as the yacht retreated from the wind; and this could be shown by the decrease in force of the *apparent* wind, as represented in Fig. 61.

This will serve to show the hopelessness of a yacht trying to beat against a 3 or 4-knot current with a very light breeze.

In sailing *before the wind*, an opposing current would have the effect of increasing the speed of the yacht *through the water*, but it does not follow that her speed over the ground would be increased. On the other hand, a favourable current might carry a vessel along over the ground faster than the wind was travelling, and she may have no motion at all through the water.

This brief description of the influence of currents on the effectiveness of sails by no means exhausts the case, but sufficient has been said to make clear the nature of the influence, and will enable yachtsmen to better understand the importance of working with a favourable current, or shunning an unfavourable one as much as possible.

It was shown on page 85 that there could be a great gain in economising distance by lying close to the wind, but the matter for consideration is this: would the gain in distance compensate for the loss in speed? As

we know what the speed per hour is on different points of sailing, this question can be answered by, for say a distance of 20 miles, a very simple calculation, the results of which are here given. Thus:

	Hrs.	Mins.		Hrs.	Mins.
3½ points from the wind*	4	0	4½ points from the wind.........	4	0
4 " " "	3	53	5 " " "	4	11

Thus, say, two cutters of 80ft. water-line each, equal in every respect as to speed, stability, and spread and effectiveness of canvas, set out to beat 20 miles to windward, and one lay 3½ points from the wind and the other 4½ points, there would be no loss or gain; but if one of them lay 4 points from the wind she would gain 7 minutes, and if she lay 5 points from the wind she would lose 11 minutes. Thus for a cutter beating to windward a course of 4 points from the real wind would appear to be the most advantageous.

It must not be supposed, if of two vessels of 80ft. water-line each, one is a schooner, and the latter makes a course of 4½ points from the wind, and the cutter 4 points, that there will be, providing their general speed and area of canvas be equal, only seven minutes' difference between them at the end of a twenty miles' thrash to windward. The schooner will probably be the equal of the cutter with the wind abeam; but it is a very different matter when the wind and the sails make a very small angle, as in close-hauled sailing. For such sailing a great portion of the sails near the upper, lower, and after edges are ineffective; consequently, the greater the number of parts a vessel's sails are in, the more edges there will be, and the greater will be the loss of propelling power; and, further, the eddied wind thrown off by the sails greatly interferes with the direct or impelling currents of wind. Beyond this, a schooner suffers in stability, inasmuch as she has to carry the weight of two masts and two sets of rigging, with loftier masts; therefore, for any given area the heeling moment of the sails of a schooner will be greater as the centre of effort will be higher; for this reason alone the effectiveness of her canvas would be inferior to that of the cutter, as it would be reduced, practically, in proportion to the co-sine of the angle of heel.

This leads us up to the point that sails are not really planes, but surfaces which are more or less concave. There is no doubt that the general *pressure* on a surface is equal, whether that surface be a plane or a concave one, providing the areas are equal; but, on surfaces like those of sails, it is not the direct pressure (such as it would be when sailing

* See Col. 2 in Table, page 94.

dead before the wind) that drives a vessel ahead; but, as previously explained, a component of the wind force which strikes the sail at some angle. The exact value of this component for a concave surface is undeterminable; but experience has taught us that it is vastly larger for flat surfaces; and that baggy sails press a yacht to leeward much more than flat sails do.

When the wind force applied to a sail comes obliquely from ahead, as in close-hauled sailing, there is a plus pressure on the fore part of the sail, and then the centre of pressure is far ahead of the centre of area, according to the intensity of the pressure and the angle of the sail. The results of some experiments which have been published would appear to indicate that the centre of pressure for close-hauled sailing is about one-third the breadth of a sail from its fore edge or luff.

The late Mr. Wm. Froude gave this subject a great deal of attention, and, in speaking of the effect of the wind on baggy sails, said:

> A striking indication of a distribution of fluid pressure on a curved surface is supplied by the *primâ facie* paradoxical curvatures into which sails often arrange themselves under the effect of wind, as is specially noticeable in jibs. In these sails the edge is gathered in, so as to form, immediately behind the rope, a narrow tapered belt of slack canvas, which becomes conspicuously bagged out by the pressure of the wind. When the wind strikes the sail obliquely from ahead, say at an angle of 45° with the line of the keel, the general wind pressure which the reaction of the rest of the sail produces swells out the "baggy" belt of canvas. As the vessel is pressed closer to the wind, it is this part of the sail which will first begin to flap or "lift."

There is no doubt that, when sails go into the "bag" described by Mr. Froude, they are very ineffective, such as old sails are, or those made of very thin light canvas, such as jib topsails, which will not stand, but are "all of a lift" in the fore part directly the wind freshens. Within certain limitations, the heavier the canvas, or the more rigid and unstretchable it can be made by narrowness of cloth or other means, the more wind will the sails usefully resolve.* Sailing masters generally well understand the importance of having the fore part of a sail flat and "unbaggable." Hence, sailing with an old mainsail, we frequently see them wetting the luff, to shrink the flax and so strain this part of the sail flatter. But in old-fashioned sails (and in some ill-cut modern ones) the after part of the sail also went into a bag, and the idea was that the wind should not be allowed to escape. But the real effect of a bag in the after part is to make a "back sail;" and, of course, a back sail retards a vessel's progress, and, in the case of after sail, helps to turn the vessel's head towards the

* The main objection to very narrow cloths is, that by multiplying the seams to obtain rigidity the frictional resistance is increased, and the most perfect canvas would be a seamless sail.

wind, by pressing her stern to leeward. The conclusion is, that the general pressure on a baggy sail is the same as on a flat surface of equal area if that pressure be applied at right angles to the plane; but if applied obliquely the component of the pressure (represented by *n x*, Fig. 62) which drives the vessel ahead is smaller, with a baggy sail, whilst the pressure that drives her to leeward, and assists in heeling her, is much larger. Before the wind this is a matter of no consequence; but by the wind it is evidently of the utmost importance that the sails should be perfectly flat, and that they should be well cut, without folds or girts of any kind, and that they should never go into bags in consequence of the canvas being soft or elastic.

It has been said that the centre of lateral resistance presents the point through which the resistance of the water to the sideway motion of a vessel acts, and the centre of effort of the sails represents the point through which the force acts which endeavours to impart sideway or broadside motion to the vessel. It is therefore evident, if these two horizontal forces do not act in the same vertical line, that some disturbance must take place in the direction of the vessel's motion. In short, the *horizontal* distance represented by *q x* or C E *s* in Fig. 57, page 83, is a coupling lever tending to turn the vessel towards the wind. When such conditions exist, a vessel requires what is known as "weather helm"; that is, if the vessel's head has a tendency to fly up in the wind, the *rudder is turned to leeward* by bringing the helm or tiller to windward. (The distance *k x* in Fig. 57 shows the length of the lever upon which the rudder acts to turn the vessel. See the next chapter.) It is obvious that if C E were directly over *x* no such lever would exist, as the force accumulated in C E would have no tendency to turn the vessel, either on or off the wind, and the vessel would "steer herself." If on the other hand *x* were at *q*, and C E at *s*, it is clear that the effort of the sails would be striving to turn the vessel's head *off* the wind, and she would in fact require "lee helm." Thus, two bad faults in a vessel whilst sailing by the wind are dependent on the fact that the centre of effort of the sails does not act in the vertical line in which is the centre of lateral resistance. It would appear to be a very simple matter to so arrange a vessel's sails that the centre of effort came over the centre of lateral resistance, presuming the latter to be determined; but, owing to the concavity of the sails, and the fact that the pressure varies considerably on them (there being always a plus pressure on the fore part or luff of the sail, as already explained), the centre of effort cannot be accurately computed. That is, it is always some distance ahead of the calculated centre; but, as the centre of lateral resistance is likewise ahead of the calculated centre (see page 69), it is found in practice both safe and useful to treat the

calculated centres as if they had been correctly determined, with the following qualification: Experience teaches us that, to obtain the largest average of advantages, the calculated centre of effort of the sails should be some distance forward of the calculated centre of lateral resistance, and this distance may vary from ·01 to ·03 of the length of the load line, the latter ratio being necessary in vessels which are deep under the mast or which have a deep fore-foot. With such a ratio as ·03 a modern cutter with her mast well forward and a very raking keel and stem would carry "lee helm," or slack helm, as it is sometimes termed, in light winds, and be slack in stays, and probably a ratio of ·02 will be a safe one to adopt. Thus, if a vessel be 50ft. on the load line, then 50 × ·02 = 1ft., which is the distance the calculated centre of effort of the lower sails is to be ahead of the calculated centre of lateral pressure on the immersed portion of the hull. If a smaller ratio be taken the vessel will in strong winds carry an injurious amount of weather helm, or be, as it is termed, too "ardent"; a little weather helm is, however, of much value, not only for effective sailing by the wind, as the resultant of the pressure of the water on the rudder assists in pressing the vessel bodily to windward; it also promotes quickness in tacking, as the ardency causes the yacht to "fly to" directly the weather tiller lines are loosened.

The foregoing, it should be understood, refers to yachts with much rockered or very raking keels. In the case of yawls it is generally found that the calculated centre of effort requires (relatively to the centre of lateral resistance) to be a little further aft than in either cutters or schooners, as the mizen is not a very effective sail on a wind, the eddy wind off the mainsail causing it to lift; also a yawl's main mast is usually farther forward than a cutter's, and it should be noted that the position of the centre of effort of the largest driving sail influences the position of the general C.E. more than the calculation shows.

As the longitudinal component ($n\ x$, Fig. 62, page 90) of the force of the wind acts through the centre of effort of the sails considerably above the centre of lateral resistance, a couple is formed (C E q, Fig. 57) tending to depress the bow; but, as the longitudinal stability of a vessel is so great, but little depression will actually take place. Thus, take the case of Seabelle at a speed at 7 knots, her resistance on the water would be about half a ton and the distance (C E q) is 50ft., and at such speed the moment would be 50ft. × 0·5 ton) equal to 25 foot tons, which would only cause Seabelle to be depressed by the head 1½in. Of course, as the resistance increased, as it would very rapidly in the case of vessels with full bows driven at high speed, the moment would increase and the bow would be further depressed. In small yachts and boats, which are made to carry sail areas (by aid of

trimming ballast up to windward) out of all proportion to their size, the depression of the bow is often very considerable, and has to be met by shifting some of the ballast or crew farther aft. It must not, however, be concluded that depression by the bow is entirely due to sail pressure; such is by no means the case, as will be shown in the chapter on "Resistance."

In apportioning sail-spread to a yacht it is usual to consider the area in relation to the speed expected, and the greater the area in proportion to the displacement and wetted surface the greater should be the speed at any given wind-force and angle of heel. The wetted surface is proportional to the square of the length, or breadth, or draught, and the displacement to the cube of either of these dimensions; hence the wetted surface will be proportional to the $\frac{2}{3}$ power of the displacement in vessels of similar form.*

This is particularly true of modern racing yachts, the dead wood being

* It should be noted that, in all cases where "vessels of similar form" are referred to, this means, strictly, that the dimensions only are altered and in exact proportion, such as would be the case if we took a drawing to an inch scale, say, and applied a half-inch scale to it, or *vice versâ*. If, for instance, we took the 20-rater Mimosa of 25·4 displacement with a wetted surface $= D^{\frac{2}{3}} \times 94$, and endeavour to find the wetted surface of Ghost, whose displacement is 30·5 tons from Mimosa's, the result would be very wide of the mark. The reason is this: Mimosa has very hollow sections, whilst the Ghost has very little of hollow. But even in vessels not so widely different in form there is not a constant ratio between $D^{\frac{2}{3}}$ to wetted surface, as may be gathered from the following table:

Yacht.		Wetted Surface. / $D^{\frac{2}{3}}$
*Mimosa	20 rating	94·0
*Arrow	90 "	86·6
*Vendetta	40 "	85·8
*May	54 "	81·8
*Florinda	120 "	78·6
*Quinque	5 "	78·6
*Vanduara	99 "	73·6
*Vreda	20 "	73·5
*Minerva	21 "	75·3
*Ghost	20 "	73·0
†Penitent	51·9ft. ...	90·0
†Faugh-a-Ballagh	30ft. ...	86·0
†Isolde	66ft. ...	86·0
†Windfall	36ft. ...	86·0
†Ailsa	100·1ft. ...	81·8

* Old rating, 1887. † New rating, 1896.

The Mimosa and Arrow have hollow sections compared with the others and more gripe. Penitent, Faugh-a-Ballagh, Isolde, Windfall, and Ailsa have, relatively to the older boats, small displacements.

trimmed away fore and aft until nothing but vessel is left. In these yachts the wetted surface is found to be about equal to displacement $^{\frac{2}{3}} \times 74$. It follows if the sail-spread is to be proportioned to the wetted surfaces that it can be apportioned by the equation: $\text{Sail}_1 = \frac{\text{Sail}}{D^{\frac{2}{3}}} \times D_1^{\frac{2}{3}}$, D and D_1 being the displacements of different vessels of similar form. The sail area for any required speed (V) is determinable from any ascertained sail area and speed thus: Sail area $= \frac{V^2 \times D^{\frac{2}{3}}}{k}$. In this equation k is obtained from the performance of some other vessel thus: $k = \frac{V^2 \times D^{\frac{2}{3}}}{\text{Sail area}}$, and is termed the co-efficient of performance. The speed obtainable from any given area of canvas will be derived from: $V = \sqrt{\frac{\text{Sail} \times k}{D^{\frac{2}{3}}}}$. In these equations it is assumed that the resistance varies as the square of the speed, which is only the case at low speeds; and, moreover, as k would vary for different vessels, the comparisons of speed by the last formula would necessarily be limited to vessels of similar form and similar condition of skin. For speeds where wave-making begins (*see* the chapter on "Resistance") V^3 might be substituted for V^2, and $\sqrt[3]{}$ for $\sqrt{}$ in $\sqrt{\frac{\text{Sail} \times k}{D^{\frac{2}{3}}}}$; but even then the formula could only be trusted for use with k derived from a performance of a similar vessel.

For these reasons very little reliance can be placed on speed formulæ in apportioning sail-spread, except by comparison with other vessels whose displacements, forms, wetted surfaces, sail areas, and performances are known. The proportion of sail area is nearly 4 square feet to 1 square foot of wetted surface in most British racing yachts; but in America the proportion is nearly as six to one. The reason of this greater sail-spread per square foot of wetted surface is that in America the prevailing winds are light, and as the classification is by actual length of water-line and not by the rating, a check on sail-spread does not exist as it does here.

For low speeds the comparison of sail area to wetted surface is a pretty sure one, as will be explained in the chapter on "Resistance." Formerly small yachts in this country had a greater proportionate sail-spread to wetted surface than large yachts had; but the sail-spread of small yachts in proportion to displacement $^{\frac{2}{3}}$ was smaller. The reason of this was that for their length the depth, weight, and stability were greater in small yachts, and consequently they, by comparison, could carry more sail. Since, however, a tax has been placed on sail the smaller yachts have been given more length of load line in ratio to their rating than larger yachts have, and their sail-spread to wetted surface appears small by comparison. In yachts of all sizes sail has been much reduced; but, at the same time, the wetted surface has also been reduced, so that the proportion of sail to wetted surface is much

about the same as it was before the rating by length and sail area was adopted.*

Yacht.		Area of Sail.	Area of Immersed Surface.	Sail Area. / Immersed Surface.
*Dolphin......	2·5 rating Y.R.A.	580 sq. ft.	195 sq. ft.	2·974
*Mimosa	20 " "	2535 "	810 "	3·130
*Ghost.........	20 " "	2577 "	699 "	3·700
*Dragon	20 " "	2632 "	670 "	3·939
*Vreda	20 " "	2641 "	695 "	3·800
*Vanduara ...	99 " "	7283 "	1892 "	3·849
*May	54 " "	5000 "	1260 "	3·968
*Castanet ...	40 " "	4070 "	1095 "	3·735
*Arrow	99 " "	6900 "	1940 "	3·556
*Miranda ...	111 " "	7700 "	2228 "	3·446
*Florinda ...	120 " "	8273 "	2200 "	3·760
*Dis............	10 " "	1660 "	476 "	3·487
*Minevra ...	21 " "	3200 "	572 "	5·594
*Genesta......	109 " "	8030 "	1940 "	4·139
*Vendetta ...	40 " "	3963 "	1175 "	3·368
†Ailsa	100·1ft.	10889 "	2373 "	4·161
†Penitent ...	51·9ft.	3061 "	774 "	3·954
†Isolde	65·7ft.	4006 "	1100 "	3·637
†Sorceress ...	24·7ft.	319 "	105 "	3·040
†May	62·9ft.	3809 "	1260 "	3·023
†Windfall ...	35·8ft.	1176 "	343 "	3·428

For equal displacements so many square feet per ton are compared, but when the displacements are unequal the comparison must be made with the two-third power of the displacement ($D^{\frac{2}{3}}$) thus:

Yacht.		Displacement.	Sail Area.	$\frac{\text{Sail}}{D}$	$\frac{\text{Sail}}{D^{\frac{2}{3}}}$
*Dolphin	2·5 rating.	3·6 tons.	580 sq. ft.	161·1	252·2
*Mimosa	20 "	25·4 "	2535 "	100·1	293·4
*Ghost	20 "	30·5 "	2577 "	84·5	265·6
*Dragon	20 "	26·1 "	2632 "	100·7	272·7
*Vreda	20 "	28·9 "	2641 "	91·4	280·5
*Vanduara	99 "	130·0 "	7283 "	56·0	283·3
*May	63 "	60·5 "	5000 "	82.6	326 2
*Minerva	21 "	21·0 "	3200 "	152·3	438·2
*Arrow	99 "	108·0 "	6900 "	64·0	305·3
*Miranda	111 "	160·0 "	7700 "	48·1	261·3
*Florinda	120 "	150 0 "	8273 "	55 1	292·3
*Genesta	109 "	141·0 "	8030 "	57·0	297·4
*Vendetta.......................	40 "	51·9 "	3963 "	76·3	289·0
†Ailsa	100·1ft.	156·0 "	10889 "	70·0	375·5
†Isolde	65·7ft.	46·0 "	4006 "	90·0	312·9
†Sorceress.......................	24·7ft.	0·66 "	319 "	483·3	425·3
†May	62·9ft.	60·5 "	3828 "	62·9	234·0
†Penitent	51 9ft.	26·8 "	3061 "	114·2	355·8
†Windfall	35·8ft.	8·2 "	1176 "	143·4	294 0

* The formula for the rating is $\frac{\text{L.W.L.} \times \text{sail area}}{6000}$ = Rating 1886 to 1895.

† The formula for the rating 1896 will be found explained on page 16, $\frac{L+B+\cdot75\,G+5\sqrt{\text{sail}}}{2}$

Another correct method of comparison is by the square root of the sail area and the cube root of the displacement (expressed in cubic feet).

In British yachts the square root of the sail area is usually found to be five times the cube root of the displacement, but in large modern racing cutters it is often six times, and this applies to American yachts also.

The displacement of Ceilsa is 156 tons, which, multiplied by 35, gives 5460 cubic feet. The cube root of 5460 is 17·6, and the square root of her sail area 104·3 then $\frac{104 \cdot 3}{17 \cdot 6} = 5 \cdot 926$, which is the relation her displacement bears to her sail-spread.

To ascertain what the sail-spread should be for any vessel in similar proportions the formula would be, in round terms,

$$\sqrt{\text{Sail area}} = (\sqrt[3]{\text{Displacement in cubic feet}} \times 6).$$

TABLE OF VELOCITY AND PRESSURE OF WINDS.

Velocity in knots per hour.	Pressure in lb. per sq. ft.	No. of force.	Description. of wind.
1.	·0067		
2.	·027	1.	Light air.
3.	·060		
4.	·107	2.	Light wind.
5.	·167		
6.	·240	3.	Light breeze.
7.	·327		
8.	·427	4.	Moderate breeze.
9.	·540		
10.	·667		
11.	·807		
12.	·960	5.	Fresh breeze.
13.	1·13		
14.	1·31		
15.	1·50		
16.	1·71		
17.	1·93	6.	Strong breeze.
18.	2·16		
19.	2·41		
20	2·67		
22.	3·23	7.	Moderate gale.
24.	3·84		
26.	4·51		
28.	5·23	8.	Fresh gale.
30.	6·00		
32.	6·83		
34.	7·71	9.	Strong gale.
36.	8·64		
38.	9·63	10.	Heavy gale.
40.	10·7		
50.	16·7	11.	Storm.
60.	24·0		
70.	32·7		
80.	42·7	12.	Hurricane.
90.	54·0		
100.	66·7		

CHAPTER VII.

THE ACTION OF THE RUDDER AND STEERING EFFICIENCY.

THE ACTION OF THE RUDDER.

A VESSEL moving through water is steered or turned by the action of a couple, the arm of which is the centre of lateral resistance and the centre of effort of the rudder (see $k\ x$, Fig. 57, page 83). The streams of water that meet the oblique surface of the rudder when it is put over represent a pressure which can be decomposed into a force acting at right angles to its surface; and it is evident that the application of this force would cause a vessel pivoted on a vertical axis through her centre of gravity to rotate about that axis, and the speed of the rotation would be dependent upon the magnitude of the force applied, the extent of area of the rudder, and the length of the arm of the couple before referred to. But a vessel is not so pivoted, and turns as it were in a ring and, at first, about an instantaneous axis, which does not pass through the centre of gravity of the ship. The following conditions dependent upon the putting over a rudder to turn a vessel are chiefly gleaned from a paper read by Dr. Woolley before the British Association.

Assume that Fig. 64, on the next page, is a yacht proceeding in the direction of the arrow A. B is the rudder put over to starboard to an angle of 35° by the tiller being pushed to port, and the arrow D represents the magnitude of a force acting upon the rudder in a direction at right angles to its surface; *a* will be regarded as the point of application of that force; but the force D can also be taken as an equal force acting in a parallel direction through the *centre of gravity* of the yacht *x*, and shifting the vessel sideways in the direction of the arrow E, combined with the couple of the force whose arm is represented by the distance the centre of effort of the rudder, *a*, and the distance the centre of the lateral resistance, *o*, are apart. The effect is, that the direct forward motion of the vessel becomes altered to one of rotation in the direction shown by the curved arrow *t* about an

instantaneous axis k, thus determined: draw $x\ q$ at right angles to D a through the centre of gravity; and draw $x\ y$ (*equal to the radius of gyration of the vessel*) at right angles to $x\ q$. Next join $q\ y$, and $y\ k$ is drawn at right angles to $q\ y$, cutting $x\ q$ in k produced; and k is the instantaneous axis. By "instantaneous axis" is meant the point on which the vessel turns upon feeling the first influences of the rudder, and this point generally lies considerably before the centre of gravity; hence it always appears that the stern of a vessel moves much faster than her head in turning; and this is really so at first, but when the vessel is kept turning the axis of rotation shifts aft until it rests in the centre of gravity of the vessel.

Components of the force in the direction E or D are employed partly in checking the vessel's way, and partly in driving her sideways nearly at right angles to her keel.* These components ultimately balance each other, and the vessel then continues to turn under the influence of the couple formed by $a\ o$ round a vertical axis passing through the centre of gravity, x.

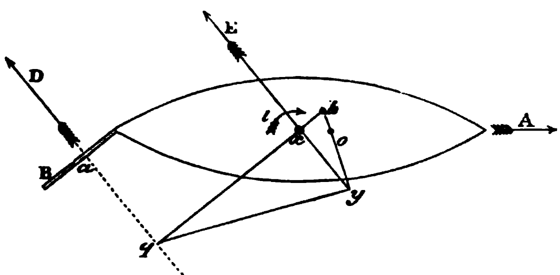


FIG. 64.

Sensibility to the helm, *i.e.*, quickness and readiness in a ship to go about, is a most important quality. At the first moment the angular acceleration, which is the measure of this sensibility, varies directly as the moment of the water-pressure on the rudder, and inversely as the product of the weight of the ship and the square of the radius of gyration about a vertical axis through the centre of gravity.

The angular acceleration or turning motion is at first very small, and its initial magnitude mainly depends upon the rapidity with which the rudder is moved so as to bring pressure on it, and upon the radius of gyration of the vessel. As the angular velocity is accelerated, so does the resistance to rotation increase until the moment of resistance balances the moment of the couple formed by the pressure on the rudder. The angular velocity, or turning motion, then becomes uniform.

* When a yacht carries weather helm the rudder is turned to leeward, and one effect of thus using the rudder is to check the forward speed and to press the vessel bodily to windward; and the latter influence may considerably lessen the leeway.

Now the moment of the water pressure on the rudder varies as the length of the couple *a o* and the area of the rudder; and the radius of gyration is dependent upon the length of the ship and the stowage of her weights in a fore and aft direction; thus a long vessel would have her already slow turning power further diminished by the stowage of weights in her ends.* If the arm of the couple on which acts the pressure on the rudder be shortened, the steering efficiency will be diminished; and it has been contended that a raking sternpost in this way shortens the arm of the couple. This in some cases is, as in that of short vessels, a mistaken contention, as generally the centre of lateral resistance is carried farther forward by the raking sternpost than is the centre of éffort of the rudder.

Beyond this, there is usually a much greater length of sternpost when it rakes, and generally the area of the rudder is thereby increased, inasmuch as the breadth only of the rudder appears to be regarded as a matter of importance, and not its depth, so far as yachts are concerned. Still undoubtedly in the case of long vessels with very raking sternposts their handiness has been improved by taking the heel of the sternpost farther aft. Some yachts with raking sternposts have enormous rudders, and, although there may be some danger in using them in the case of sternway and in scending, there is no doubt they are efficient. But a rudder hung on a raking sternpost is not wholly effective, inasmuch as a component of the pressure on it is exerted in a vertical direction, and tends to drag the vessel's stern under. This can only be regarded as a disadvantage, and a further disadvantage is that the rudder is difficult to put over, as it has to be lifted every time; but the latter difficulty is overcome by making the tiller longer than would be required for a similar rudder hung on an upright sternpost.

With regard to resistance to rotation, this mainly depends upon the area of the immersed longitudinal section, and particularly upon the amount of dead wood fore and aft. By reducing the dead wood fore and aft the resistance is proportionately decreased, and, moreover, the radius of gyration would be somewhat shortened by the reduction of the fore and aft weight; but almost equal effects would be produced by taking away from the dead wood forward and aft, and concentrating it in the middle of a vessel under the keel. The effective surface for lateral resistance

* The bad effect of weights in the "ends" of small boats is very noticeable, and we have known cases where a boat has been cured of a tendency to miss-stays by simply concentrating her weights or ballast amidships. On the other hand, a very old practice to ensure a small boat staying is for someone to get into the bow as the helm is put down; but this was to lighten aft and depress forward, so as to make the boat turn on her fore foot. The bow, however, should be relieved of the weight directly the boat is head to wind, or she may fail to "fill" or fall off.

would be maintained, and the radius of gyration would be still shortened.*

To sum up, the quickness of a vessel in answering her helm and the smallness of the circle in which she will turn depend:

1. Upon the smallness of the weight of the vessel and her radius of gyration.

2. Upon the area of the rudder, and the length of the couple upon which it acts, and upon the time it takes to put the rudder over.

3. Upon the area and form of the immersed longitudinal-vertical section of the vessel.

The double-boarded boat, as depicted in Fig. 65, affords peculiar advantages for lengthening the arm of the turning couple, as by lifting the board (h) the centre of lateral resistance is thrown very much forward, and the area of dead wood at the after end (which might be necessary in

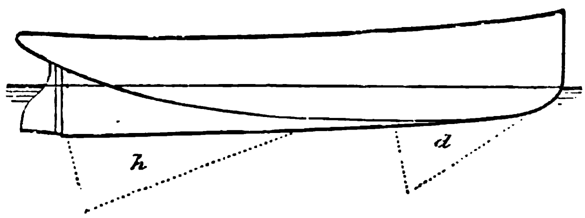


FIG. 65.

a sailing vessel to check leeway or to balance sails) is considerably reduced. In tacking, if the vessel's "way" were stopped as she came head to wind, and the rudder thereby became useless, the fore board d could be raised and the after board lowered, and thus the head of the vessel, by aid of the fore sails, would readily fall off the wind.

All kinds of fanciful forms have been given to rudders, and a practice came in a few years ago of putting the greatest breadth near the surface of

* Light centre-board yachts are frequently said to be more sluggish in stays than keel yachts, and no doubt there is some truth in this, although, looking to their immersed form alone, they ought undoubtedly to turn quicker than keel yachts; but we are inclined to think that the superiority of the keel yacht in this respect is owing to her greater momentum due to her greater weight, so that she carried her way until well round on the other tack. If two vessels are to be impelled by an equal force at equal speeds, and one vessel is heavier than the other, then it will take a longer time to get the heavier vessel up to the required speed than it would the lighter vessel; so also when the force is withdrawn it will take a longer time to exhaust the momentum due to the speed of the heavier vessel than it would the momentum of the lighter vessel. Thus heavy vessels (although they may be turning in a circle of greater radius) are said to "shoot" far in stays, and fill on the other tack without losing way. The assumed advantages which a heavy vessel has in this respect will be more apparent in strong winds when the momentum, due to high speeds, is very great; the case of a steam vessel which has continuous propulsion, whilst turning is of course different.

the water. We believe this practice was owing to a vessel once having the lower half of her rudder accidentally carried away, and the subsequent report that she steered better with the part than with the whole. We are inclined to think that in this case the inefficiency of the entire rudder depended on its being too big for the crew to use; at any rate, an experiment made by Mr. Froude with a model of H.M.S. Encounter clearly proved that the lower half of a rudder is more effective than the upper half. The rudder was in two parts, *i.e.*, squares of equal dimensions, and it was found that the upper half required to be put over to 20° to balance the lower half at 10°, in order that the vessel might follow a straight line. In the case of a very raking sternpost there may be a small advantage in having the broadest part of the rudder in the top half, as its centre of effort would thereby be carried farther aft; but generally there can be no doubt that the deeper the main area of the rudder is immersed, clear of the fulness of the after body, the more effective it is. As a clean

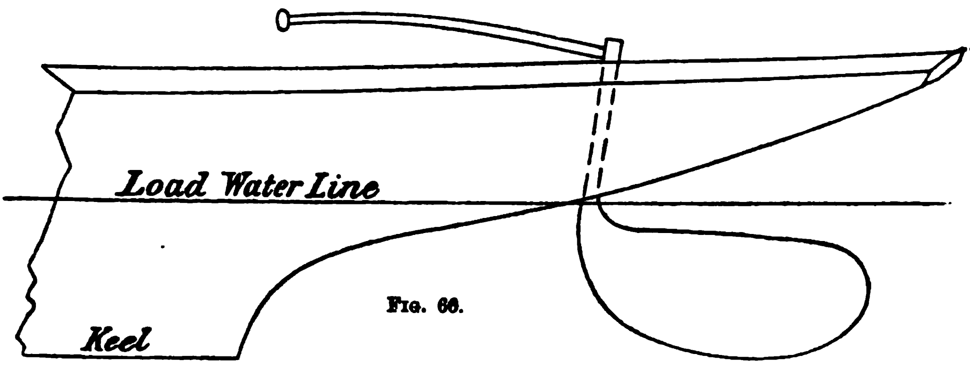


FIG. 66.

run aft is a great assistance to the effectiveness of the rudder, it is quite possible, if an experiment similar to that tried on the Encounter were tried with a yacht, that a less difference in the effectiveness of the two halves might be found. The advantage of keeping the rudder well immersed—both in smooth and disturbed water—is so well understood by small boat sailers that we find many small and shallow craft with their rudders dropping considerably below the keel. The only disadvantage of this arrangement is that, if the boat carried much weather helm, and if the centre of effort of the rudder were much below the centre of lateral resistance, the pressure on the rudder would tend to increase the boat's heel, although not to a considerable extent. (These rudders are so hung that in shallow water they lift without unshipping.)

In the other small craft, and in torpedo boats and steam launches, the whole of the dead wood aft is frequently cut away and the rudder hung free under the stern like the blade of an oar (see Fig. 66). It was

formerly thought that a rudder's action was increased by the dead wood, but it is difficult to see how this could be; at any rate, the removal of the dead wood aft more than compensates for its presence by allowing the stern to move readily away in the direction of the arrows D and E (Fig. 64).

There appears to be no definite rule for determining the breadth or area of rudders in sailing yachts, but in proportion to the area for lateral resistance it is much greater in small yachts than in large, for the reason that there is a limit to the size of rudder a helmsman can handle. Beyond this, large rudders are highly dangerous, especially in the case of making sternboards and in scending in a heavy sea.

In yachts up to 40ft. length of load-line the area of rudder may be found as much as $\frac{1}{15}$ the area for lateral resistance, but in a yacht of 80ft. length it is seldom more than $\frac{1}{30}$ of that area; and in large schooner yachts no more than $\frac{1}{40}$. The width of rudders to some extent is governed by the depth of a vessel, and will be found to vary from $\frac{1}{15}$ the length of L.W.L. in small vessels, to $\frac{1}{20}$ to $\frac{1}{25}$ in large, according to the shape of the rudder. If the rudder is much rounded away the breadth is increased in order to obtain the necessary area. The area of rudders of steam yachts is somewhat less than in sailing yachts, and the breadth is largely governed by the draught of water.

With regard to the relative efficiency of broad and narrow rudders, it appears, from experiments made for the Admiralty some years ago, that a rudder of, say, 3ft. in breadth, put over to an angle of 30°, would have double the efficiency or turning power of one 6ft. in breadth put over to half the angle, or 15°; and the force required to move the rudder would be the same in either case. Thus there can be no increase in the efficiency of a rudder by the mere addition of breadth without an increase in the power to use it, and a very large rudder, if used so as to obtain its greatest efficiency—which is found to be when put over to an angle of about 35°—means a great retardation of speed; or, the smaller the circle a vessel is made to turn in, the more speed will be retarded. A variety of steering gear has from time to time been introduced, which, by the aid of the wheel, has enabled one man to do the work of two or three. Of these contrivances Jarman's is the one most generally used; but, so far as racing yachts are concerned, we think a long tiller and relieving tackle are to be preferred.

As a rule no calculation is made in fitting a vessel with a tiller, although but roughly it is generally about one-tenth the length on load line; if the rudder cannot be readily used with the tiller, extra tackles are fitted to it, or the tiller is discarded and a longer one used in its place. However, a

very simple calculation will approximately show the force that would be required at the tiller head for any given area and angle of rudder, length of tiller and speed. At a speed of 6 knots the resistance of a plane moved at right angles to its surface is 112lb. per square foot, and the resistance increases as the square of the speed; also, the resistance to a plane moved obliquely in water varies nearly as the sine of the angle of inclination. Then, say that the rudder had an area of 30 square feet, and put over to an angle of 30°; then $112 \times 30 \times$ (0·5 sine of angle) $= 1680$lb. That is, the pressure on the rudder put over 30° whilst the vessel is moving at a continuous speed of 6 knots, will be 1680lb. This force of 1680lb. is exerted on a lever, the length of which is the distance the centre of pressure on the rudder is from the sternpost. The centre of pressure on planes moved obliquely in fluids is not at the geometrical centres, and experiments have shown that the resultant of pressure on a rudder, when inclined to 30°, is at or near one-third the breadth from the anterior edge, or edge next the sternpost. This has been well proved by "balanced" rudders pivoted at or near one-third their breadth from their fore edge; no power is required to put the rudder, so pivoted, over beyond that necessary to move the water, and to overcome the friction of pintles, rudder post, and the effort that would be required to move the rudder if the vessel were not in the water.

If the rudder had an average breadth of 3ft., the resultant of pressure on it would thus be 1ft. from its edge next the sternpost, and the length of lever on which the pressure on the rudder acted would therefore be 1ft.; and 1680×1ft. $= 1680$ foot-pounds which is the moment to be overcome by a force at the tiller head. The magnitude of this required force can be calculated: thus, say the tiller is 8ft. long, then the force required at its head to balance the moment of the rudder, when put over to 30°, will be thus found:

$$\frac{1680 \times 1}{8} = 210\text{lb.}$$

That is to say, a steady pressure of 210lb. would be required at the tiller head to keep the rudder over at 30° if the vessel were moving at a continuous speed of 6 knots.

All other things being equal, the diameters of the circles vessels will make in turning are in direct ratio to their dimensions. Thus, take two yachts, one of 25 tons and the other 200 tons: their length would be 50ft. and 100ft., and the yacht of 50ft. should in turning describe a circle of just half the diameter of the one 100ft. long. The latter will describe a circle in turning of about twice her own length, or 200ft. in diameter (assuming her to be under continuous steam power, with her helm over to 35°), at a speed of eight knots, in about two minutes, as the direct speed would be

retarded nearly two-thirds. Under sail, however, and in getting from one tack to the other, a yacht does not describe a circle, but only one-fourth, or the arc of a quadrant, if she lies four points from the wind; and presuming a yacht in tacking traversed the whole arc of a quadrant from the force of a direct speed of eight knots an hour, she would be a little

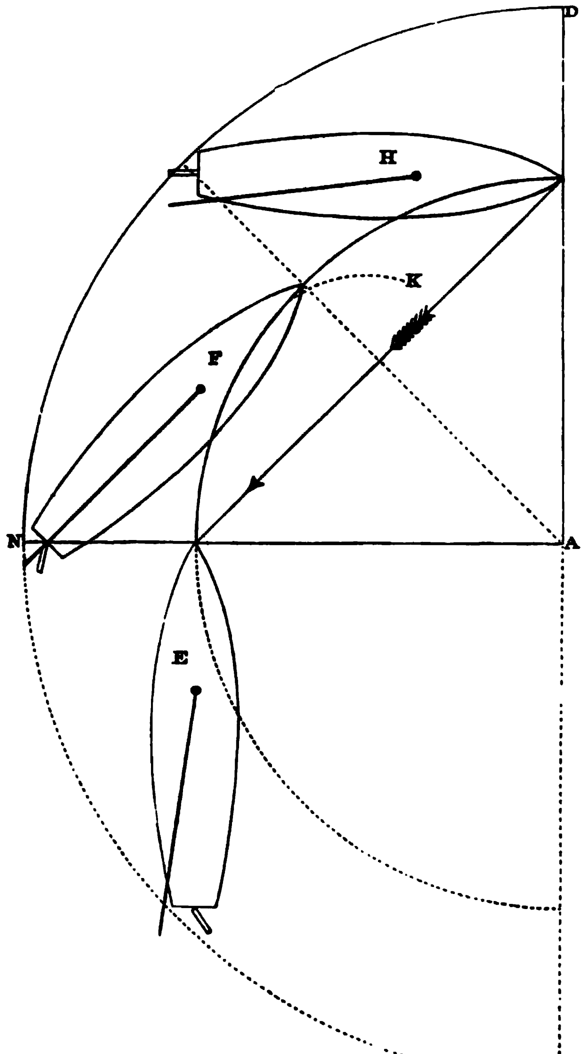


FIG. 67.

under half a minute in getting from one tack to the other. For the purpose of illustration, we can assume that a yacht maintains uniform speed whilst turning. In Fig. 67 A N D form a quadrant, and let E be a yacht proceeding in a direction parallel to B A D, with the wind blowing as shown by the arrow, four compass points or 45° from her course. The yacht when

in the position F would be head to wind, and when at H would be on the other (port) tack, on a course at right angles to that at E. But the speed of the yacht would diminish from the time her helm was put down, and when she arrived at F, head to wind, her propelling power would be gone entirely. She would proceed a little farther on the arc of the quadrant under the influence of her rudder, but would eventually pay off, under the action of her head sails, and come fairly on the other tack (proceeding in a direction parallel to A N), somewhat in the direction K.

As a matter of fact, the portion of the circle which a yacht describes in tacking is always of greater radius than her own length, or the circle in diameter is greater than twice that length. The helm cannot (and frequently it would be inadvisable so to do) be put over to 35° suddenly in large yachts, and generally the yacht will be head to wind before the helm is so over to 35°. Thus in a large yacht, say of 100 tons, the helm cannot very well be put down too quickly, or indeed even quickly enough. On the other hand, in small vessels, the helm may be put over too suddenly, and, by forcing the yacht to describe a segment of a circle of very small radius, her way becomes deadened; so that, when she gets on the other tack, she rests motionless in irons, instead of springing off with way almost unchecked. This is a very important matter, as successful tacking does not depend upon merely getting from one tack to the other, but in getting *off* on the other tack without losing way. The helmsman of a large craft cannot easily make a blunder, as generally the speed with which he can walk a tiller down to leeward is that most suitable for preserving continuous way in tacking. This is not the case in small vessels, as the tiller can be put over with one hand suddenly, and the vessel brought head to wind (when, of course, her propelling power is nearly gone) in from three to five seconds. Now it is obvious that some turning power is required after a vessel is head to wind, as at that point she is only half-way towards getting on the other tack, and no help can be obtained from the rudder if there be no way on the vessel. Thus, if a vessel's way be stopped before she be fairly on the other tack, the head sails will have to be kept a-weather; but in smooth water at least, and in a whole-sail breeze, a skilful helmsman will never require such aid, but will tack his vessel fairly by the influence of the rudder, the head sheets being lightened up of course; but not before the luff or fore edges of the sails begin to lift, and as a rule the jib sheet should be the first to be eased, as it will be the first to lift when the yacht is about half way towards the head to wind point. The sail will no longer be of use as a propelling force, and what little wind it holds will now only tend to check the vessel coming into the wind. As she begins to pay off, the head sails would be sheeted to the marks.

It should not be lost sight of that in performing the first half of the operation of tacking—that is, bringing a vessel head to wind—the assistance given by the sails to the vessel in turning does not retard or deaden the vessel's way like the action of the rudder. By letting fly the head sheets at the right moment, the centre of effort of the other sails would be thrown so far aft that a very long area would be formed by the couple C E *s* (see Fig. 57, p. 83). Also if a vessel carries a fair amount of weather helm it will be easy to tack a vessel by working the sheets alone without actually putting the helm down. Directly the tiller is released (or the rudder pressure which had been used to keep her out of the wind removed) the vessel would fly head to wind, the jib sheet being, of course, also let go, and, with the fore sheet kept a-weather, her head would soon pay off; it might, however, in rough water, be found necessary to ease the main sheet before she would gather way, as the foresail being aback would, assisted by the diving, tend to put stern way on. However, so far as small vessels of 40 tons and under are concerned, they fly round so quickly in smooth water that there is barely time to get their head sheets over before they are drawing ahead on the other tack. At the Institution of Naval Architects in 1871 the late Sir E. Belcher, speaking on the handiness of sailing ships, stated that when he was in charge of the Samarang he never allowed the helm to be put down in tacking. The helm was let go (which means that the ship carried weather helm) and she came head to wind, the head yards were braced aback, and thus the Samarang was always got round. In construing these remarks it must not be assumed that a vessel will not stay unless her head sheets are let go; this is by no means the case, and a modern yacht will even stay against her fore sheet a-weather; the general principle of the influence of sails in tacking has only been kept in view.

The action of the rudder is much interfered with by the heeling of the vessel, and may have its effect diminished from this cause. However, the main difficulty to deal with in sailing at large angles of heel is the tendency which deep-bodied vessels with short after bodies and steep buttock lines have of "running off the helm" as it is termed; or, in other words, the tendency of the vessel to turn her head to leeward. This tendency is generally observable at the high speeds attainable with a beam or quarter wind, and mainly arises from a deficiency of pressure on the weather quarter, relative to the pressure on the lee quarter, allowing the stern to turn or lift to windward. At high speeds a large wave trough is developed on the weather side, and is the more or less pronounced as the vessel is more or less deep in the bilge or middle body. The faster the vessel is driven the farther this trough will appear aft with a crest under the quarter; and also the shorter the after body is, the nearer the trough

will be to the counter or free end of the vessel aft, and a great part of the pressure from the wave crest will be lost. As the pressure is diminished on the weather side, there is generally also an increase of pressure on the lee side, in consequence of the full quarter and counter becoming submerged. The final result is, that the vessel "runs off her helm," and unless the tiller be put to leeward in sufficient time to check the first symptom of the tendency, there will be great difficulty in bringing the vessel to. This does not, perhaps, include all the influences which are at work in causing a vessel to "run off her helm," but the main influence is as described, and it is seldom apparent except in vessels of deep form, as shallower craft do not make such large waves to windward, and, moreover, are never sailed at such great angles of heel. This tendency of "running off the helm," should not be confused with lee helm, nor with the yawing incidental to sailing in a sea-way, or to the varying action of the wind on the sails, such, for instance, as when a vessel is before the wind.

The rudder action of vessels propelled by screws is attended by some anomalies which cannot always be accurately pre-determined. In the first place, the action of a propeller on a ship whose helm is amidships tends to turn her; that is, a right-handed screw will turn the ship's head to starboard, and a left-handed one to port; and from experiments made with the Great Britain in 1845 it was known that a propeller, if suddenly reversed, would tend to turn a ship against her helm until her way was stopped. In spite of this knowledge, much attention does not appear to have been given to the subject until the year 1877, when several accidents were clearly traceable to the fact that the vessels under port or starboard helm, whilst their screws were being reversed to deaden their way, turned the opposite way to that expected. Professor Osborne Reynolds gave the subject much attention, and at last, in 1877, succeeded in getting a committee appointed, consisting of hinself, Mr. James R. Napier, Sir William Thomson (now Lord Kelvin), and Mr. W. Froude, to inquire into the matter. The committee made experiments with the Duke of Argyll's steam yacht Columba, the Earl of Glasgow's steam yacht Valetta, and several large steamers in the merchant navy.

The result of the experiments bore out all that was previously known or suspected of the action of the screw on the steering. With the steamer going nine or ten knots ahead, the engines were suddenly stopped and reversed; simultaneously with the reversal of the engines, the helm was put hard-a-port, the vessel, of course, still forging ahead. According to what is usual, with the helm hard-a-port the ship's head should have turned to starboard; but, instead of doing this, her head went off to port twenty-eight degrees, or about two and a half compass points, before the vessel's

way was stopped. The ship was then turned full speed ahead again, and when she had full way on the engines were stopped and reversed as before, but the helm was put hard-a-starboard. Between the moment of reversing the engines and that of the ship's way being stopped, the ship turned forty degrees to starboard, or nearly four points, or nearly one eighth of a circle. The experiment was tried over and over again, and the invariable result was that, during the interval of reversing the engines and the stoppage of the ship, the rudder acted just contrary to the way expected.

A fact hitherto not generally known was brought to light, that the effect of the screw on steering is largely governed by its immersion, and that its influence is greatest when it is near the surface "churning" the water. When deeply immersed it has little or no effect of itself in turning a ship one way or the other, and never under the most favourable circumstances—that is, with the screw near the surface, and a low speed—could the ship be got round in circle less than double the radius of the one she would describe under the influence of her rudder with the engines turning ahead.

Also it was clearly proved that an experiment is required for every individual vessel, and for varying speeds; thus, one steam yacht, with helm hard one way or the other, may keep a straight or nearly straight course whilst her engines are reversed and before she stops, and another may turn in direct opposition to the way she should turn under the ordinary influence of a rudder. It is thus necessary that a simple experiment should be made for every screw steam yacht, so that her master might be acquainted with her behaviour as to steering whilst her engines are being reversed. It can, perhaps, be said that a vessel turning through an arc of twenty or forty degrees cannot be of much consequence when her way is being stopped; and it might not be if it was clearly understood which way she was going to turn. It might, however, be of the utmost consequence, if the vessel whilst losing her way turned thirty degrees to port, instead of thirty degrees to starboard. This would practically be a difference of sixty degrees, or two-thirds of a quadrant, and would be quite sufficient to bring about a disaster.

One other result of the experiments was to make patent what was not very generally known, that the reversal of the engines of a screw steamer has but inconsiderable effect on stopping her way; and that the distance required to bring a screw steamer to rest mainly depends upon her size, weight, form, and speed. The distance may roughly be put down as five or six times her own length.

These peculiarities should be well known and be well considered by all in charge of screw steamers, as it is quite plain that "full speed astern" may often bring about a collision, whereas steaming ahead and using the rudder might avoid it. This may be especially the case if a steamer is

approaching another, or approaching a shore, in an oblique direction. Say that the ship or shore bears on her port bow six or seven lengths off, and by continuing she would strike the ship, or in the case of the shore go aground; if she ported her helm and reversed, she might be carried stem on to the very object she hoped to avoid; whereas by porting her helm and steaming ahead she might have cleared it, or by only stopping the engines, and not reversing them, she might have gone clear.

A screw steamer will generally turn in a circle whose radius is about four times her own length, and stop in a distance equal to about six times her own length: thus far, if the object to be avoided were six lengths off, it could be avoided by turning under full speed. This, however, supposes that the steamer can be given her full rudder power (about thirty-five degrees) at once; whereas the fact is that it takes a very considerable time to get a rudder hard over when a steamer is at full speed, and very seldom is there power enough at the helm—perhaps only one man—at the moment to do it. Of course, by stopping the engines the power gained by the helmsman over the rudder would be increased; and generally it would appear to be the wisest plan, when a screw ship gets into such a position that it is almost certain she will strike some object ahead, for her to stop her engines and put the helm over just as if the engines were still turning ahead. Of course, the engines could be reversed and the helm put the other way, to assist the effect of the backward motion of the screw on the turning; but, unless the person in charge is thoroughly acquainted with the behaviour of the ship under all conditions, simply stopping the engines and making as much use of the rudder as possible would appear to be the wisest course, as, if the screw is not revolving, the vessel will, of course, steer like an ordinary sailing ship.

As before said, very few steam yachts behave exactly the same under the influence of the rudder during stern way, or under the influence of the screw alone with the rudder amidships; and it will be found incumbent in most cases to make an experiment with every steamer in order to discover her peculiarities. In the first place, it should be ascertained how far she will go before losing way when the engines are stopped from full speed ahead, and from half speed ahead; also the distance she will traverse before losing way whilst assisted by her engines being reversed. Next it should be ascertained how many compass points the vessel will turn before losing way; and how many points she will turn and in what direction, before losing way, whilst the engines are being reversed. Also, the steering should be thoroughly tested whilst the steamer has stern way on.

CHAPTER VIII.

RESISTANCE TO VESSELS MOVING IN WATER.

UNTIL about the year 1852 it was generally supposed that a vessel, on being impelled through the water, encountered some kind of direct head resistance, the measure of which was the area of the greatest transverse section and the velocity of the vessel moved. The investigations, however, made by the late Professor Rankine, and more recently those made by Mr. W. Froude, show that no such direct head resistance can exist—that is, no resistance analogous to that which a board would meet with on being moved as a vertical plane through the water. If, however, vessels of different extreme dimensions are of similar form, then their respective midship section areas might be a just means of comparison, but only for the same reason that length or breadth would be just elements for comparison in vessels whose forms and proportions were constant.

There are two principal sources of resistance to deal with; the first is that known as "surface friction," "skin friction," or "skin resistance;" and the second is the resistance due to the formation of waves; another resistance due to the making of eddies is of minor importance.

The eddies are usually created round the stern and stem; but yachts are so formed that the eddies in the wake are unimportant, as the water re-unites astern with scarcely perceptible eddy commotion. With regard to eddies at the stem, they are as little likely to be observable; but a very thick or broad stem would, by reason of its large direct resistance, meet with such eddy resistance as well, the reason being that the continuity of the stream lines would be broken up. This was called by the late Professor Rankine "the distortion of particles of water."

This resistance would be equal to the resistance offered by the particles of water to separation and to the different influences of frictional action the particles of water undergo in passing each other. Water according to its density has the properties of viscosity, and stiffness, and resistance due to this viscosity will be more apparent in salt water than in fresh or river water.

SURFACE FRICTION RESISTANCE.

Surface friction is caused by the adhesion of the water to the immersed surface of the vessel, and it exists at all speeds; its quality is dependent upon the smoothness or roughness of the surface, and within certain limitations (the friction gradually decreasing as the after end of the vessel is approached) to the length of the surface acted upon.

Referring to the fact that frictional resistance, up to certain lengths of body, is graduated and is greatest at the entrance, Mr. Froude remarked that such might be expected, because the fore part of the body must give motion to the water in the direction in which itself is travelling, and "consequently the portion of surface which succeeds the first will be rubbing, not against stationary water, but against water partially moving in its own direction, and cannot therefore experience so much resistance from it." However, with bodies of 50ft. and upwards in length it is safe to treat the frictional resistance as equal all over the surface, and this is commonly done for over 30ft. lengths.

Upon a body 50ft. in length, moved at a velocity equal to 6 knots an hour, Mr. Froude found that the mean frictional resistance of clean copper sheathing is ·246lb. (or in round numbers ¼lb.) per square foot of wetted surface; for a blackleaded surface ·248lb.; for smooth varnished surface, ·250lb.; for Jesty's or other equally good compositions, ·250lb. For lengths over 50ft. the mean resistance would be somewhat less, but as the ratio of decrease is so small for lengths exceeding 50ft., the values given can be taken as constant. The friction resistance, on the surfaces just quoted, increases nearly with the square of the speed, or actually as the 1·83 power of the speed; but in the case of very rough surfaces, such as a bottom fouled by barnacles, weeds, &c., the increase may reach the cube of the speed.

At six knots the variation with the length of the intensity of the frictional resistance is as follows: (The resistance is expressed in fractions of a pound per square foot.)

Length of Board.		2ft.	8ft.	20ft.	50ft.
Bright copper		·410	·325	·278	·250
Foul copper		·900	·625	·534	·488
Power of speed to which resistance is proportional	Bright copper	$Speed^2$	$Speed^{1\cdot83}$	$Speed^{1\cdot83}$	$Speed^{1\cdot83}$
	Foul copper	$Speed^2$	$Speed^2$	$Speed^2$	$Speed^2$

The foul copper was a surface covered with a coarse shore sand such as used for cement stucco work, and fairly represents an ordinary

foul bottom; but, as before said, with a bottom thickly fouled with barnacles, say, as large as hemp seed, the resistance may reach nearly 1lb. per square foot, and vary nearly as the cube of the speed.

For ordinary calculations for bodies of a load water-line length of 30ft. and upwards it will be sufficient to take it as a fact that the resistance for a copper bottom increases as the square of the speed; or say it is ·25lb. per square foot at six knots, then for other speeds it will be as follows:

Speed in Knots.	Resistance in lb. per Square Foot.	Speed in Knots.	Resistance in lb. per Square Foot.
5·0	·170	10·5	·770
5·5	·208	11·0	·845
6·0	·250	11·5	·921
6·5	·294	12·0	1·000
7·0	·340	12·5	1·083
7·	·390	13·0	1·171
8·0	·444	13·5	1·263
8·5	·500	14·0	1·360
9 0	·560	14·5	1·458
9·5	·626	15·0	1·562
10·0	·700	15·5	1·670

That is to say in round numbers at 6 knots it is ¼lb. per square foot; at 8½ knots it is ½lb.; at 10½ knots it is ¾lb.; and at 12 knots 1lb., and so on.

It is worthy of note that for any given area the frictional resistance is somewhat less for a narrow and deep model than for a broad and shallow model which has a broad angle of entrance.

WAVE MAKING: THE WAVE LINE THEORY.

The resistance due to the formation of waves is involved in great complexity, and no perfectly satisfactory solution of the whole problem has yet been given to the world. However, this much is known, that until a speed is attained, when the formation of waves is appreciable, nearly the whole of the resistance up to the attainment of that speed is produced by surface friction alone.

Mr. Scott Russell was the first to define the important part wave-making plays in the resistance met with by vessels moving in water; and although his theory, as to the creation of waves by a vessel so moving, is not now accepted as a true solution of the phenomenon, yet there is abundant evidence of the correctness of the fundamental part of his theory, that there is a length of entrance and run adapted for any given speed; and if that given speed be exceeded the resistance due to wave making will be very considerable.

The wave making involved in Mr. Scott Russell's theory consisted of a bow wave and stern wave, with supplementary following waves astern in the wake of the vessel moving.

He described the wave at the bow as "solitary," travelling on in front, so long as the vessel keeps moving, in an unbroken form, when the speed is moderate, and with the vessel's bow partly in it. Mr. Scott Russell described it as the "wave of translation," or wave of displacement, and assumed and defined its properties. In outline he found it formed a curve of versed sines, and that it will only attain its natural velocity due to its length in water of certain depths, and the speed of the vessel will suffer if she be driven in shallow water at greater velocities than the wave can naturally travel in such water. This is a fact which may be constantly observed when steamboats are navigating comparatively shallow rivers the wave formation becoming much distorted and seriously diminishing the speed. Subjoined is Mr. Scott Russell's table of the velocity and length of the bow wave in given depths of water.

Depth of Water.	Speed in Knots.	Speed in Statute Miles.	Length of Wave.	Depth of Water.	Speed in Knots.	Speed in Statute Miles.	Length of Wave.
FT.			FT.	FT.			FT.
0·06	0·87	1	0·42	8·08	9.55	11	50·82
0·33	1·74	2	1 68	9·50	10·42	12	60'48
0·60	2·60	3	3·78	11·33	11·29	13	70·98
1·07	3·47	4	6·72	13·20	12·16	14	82·32
1·64	4·34	5	10·50	15·00	13·03	15	94·50
2·33	5·21	6	15·12	17·00	13·90	16	107·54
3·25	6·08	7	20·58	19·25	14·76	17	121·38
4·25	6·95	8	26·88	21·75	15·63	18	136 08
5·42	7·82	9	34·02	24 12	16·50	13	151·62
6·67	8·68	10	42·00	26·50	17·40	20	168·00

In water deeper than 26ft. any bow wave possible for a ship to make would travel at its natural velocity.

When Mr. Scott Russell made the discovery that the bow wave formed in outline a curve of versed sines, he very plausibly made it an essential part of his theory that the water lines of the bow (the outlines of the horizontal sections) should form curves of versed sines, or wave lines; also,* such a bow he declared to be the "form of least resistance," and he supported his view of the case by numerous experiments with models of variously formed bows, and the result of his experiments led him to the conclusion that a curve of versed sines is the line of least resistance.†

* *See* the "Transactions of the Institution of Naval Architects" for the years 1860, 1861, and 1879.

† To construct a curve of versed sines (see Fig. 68), divide the length, *a c*, into a number of equal parts by the ordinates 1, 2, 3, 4, 5, 6, 7, 8; make a semi-circle with a diameter, *a b*, equal to half the breadth of the vessel on the load water-line; divide the semi-circle into a number of equal angular intervals radiating from *s*, equal to the number of ordinates; from the latter

He argued, and supported his arguments by most ingenious illustrations, that the wave line bow pushes the water away as it advances in a manner which tends to produce on the water's surface the exact versed sine wave of translation. The vessel, he insisted, pushed away progressively layers of water, and as there is no room for these layers in the surrounding element abreast or below they come to the surface in the form of a wave spread over a large area, and corresponding in form to the order in which they were pushed away at different points in the bow. If, however, the bow has not wave lines, but convex, and wedge-shaped full vertical sections, the blows from the bow instead of being delivered in a more or less horizontal direction are given in a more or less downward direction, and the water surrounding the bow is thus forced down on an imaginary plane, which will not yield because it is "full of water below." The water which is struck by the bow, finding it cannot get "below," turns, as it were, on the imaginary plane, or is deflected in an upward direction and ultimately finds its way to the surface in an ill-shapened mass not at all resembling the "wave of translation;" there is, therefore a great waste of energy.

divisions, at their intersection with the half circle, produce lines parallel to *a c* (which can be taken as the line representing the length of the forebody on the load water-line); the intersections of these lines with the ordinates 1, 2, 3, 4, 5, 6, 7, will be points in the curve.

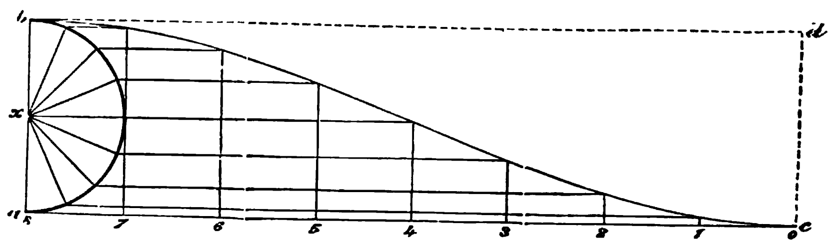


FIG. 68.

As a curve of versed sines is constant for all lengths or diameters, the ordinates 1, 2, 3 bear a constant ratio to *a b*, and that ratio is as set out in the table. The half-breadth, *a b*, No. 8,

1	2	3	4	5	6	7	8
·03806	·14644	·30866	·50000	·69134	·85355	·96194	1·00000

will be multiplied by the factors, and the products will be the lengths of the respective ordinates 1, 2, 3, 4, 5, 6, 7.

It will be seen that No. 4 ordinate is equal to half the diameter of the semi-circle, or one half of No. 8 ordinate; and if another figure similar to Fig. 68 were constructed, and the pair laid together, it would be found that the convex part of one curve won'd exactly fit into the concave part of the other. Thus, it is plain that the space inclosed within the curve *a b c* is exactly one-half the parallelogram *a b c d*, and the curve divides the parallelogram as equally as a diagonal would drawn from *b* to *c*. The co-efficient of fineness for a curve of versed sines is therefore ·5.

It must, however, be understood it is not contended that *actual water* which comes in contact with the ship is thrust ahead, but that it communicates continuous pressure on the infinite surrounding particles, and the result is the formation and sustenance of the wave as described. If the bow is full or convex, this wave will accumulate to great height immediately in front of the vessel; but if the bow be fine and well formed, it will spread over a great surface, and be scarcely discernable at speeds about equal to the square root of the length of the vessel.

It must not be supposed that either the actual character of the wave described by Mr. Scott Russell, or its creation, was generally accepted as correct. So far as the creation of the wave goes, it has been pointed out that if, as Mr. Scott Russell says, it is an absolute necessity or condition of a vessel's progress that a wave should be driven up on the surrounding surface as described, then it would be impossible to move a vessel at all if she were deeply submerged, or if she were placed in a tank full of water with a closed top. In this dilemma we fall back upon the "stream line theory" for a solution of the problem of wave creation. With regard to any particular form of water-line influencing the form of waves, the late Professor Rankine pointed out that the course taken by particles of water in gliding over the bottom of a vessel are neither over the horizontal water-lines, nor vertical bow and buttock lines, but are intermediate in position between these lines, and, in well-shaped vessels, approximate to the lines* of shortest distance, such as are followed by an originally straight strake of plank when bent to fit the shape of a vessel. Thus what is known as a diagonal or ribband line would more nearly approach the direction from which thrust is imparted to the water than horizontal water-lines.

We will next briefly glance at Mr. Scott Russell's theory as to the run of the vessel. The old theory was that some kind of suction existed, which tended to draw the vessel back; but this notion has long been exploded by the ascertained fact that the replacing water exerts a forward pressure on the hull, and it is of the utmost importance that the after body should be so formed as to admit of a maximum of this pressure. From the experiments made by Mr. Scott Russell, he concluded that the water, in refilling the cavity caused by the advance of the vessel, tended to form itself into an ordinary ocean oscillating wave of the cycloid or trochoid class, which alters in form according as the height of the wave bears a greater or less ratio to its length.

* These lines were termed "dividing lines" by Lord Robert Montagu in his treatise on "Naval Architecture," published in 1852. He strongly urged the importance of designing ships by "dividing lines" instead of by the then usual method of employing water lines only.

In speaking of the formation of this wave of replacement, Mr. Scott Russell said, "not only is there a current from *before* and another from *behind*, but a rapid current comes into the hollow from below and the *sides*; they altogether make the following wave, the refilling wave, or replacing wave. I found also that it depends upon circumstances whether the particles move in a vertical or horizontal direction. In a channel of very small depth the particles are obliged to move horizontally into the hollow. In deep water the greatest mass of water nearest to the place to which it has to come is below the vessel. With a vessel in very deep water, then, the after lines must be vertical." That is, the cycloidal form must be shown in the buttock lines rather than in the water lines. This would give sharp V-shaped vertical sections such as yachts have; and Mr. R. E. Froude says* that the results of the numerous experiments conducted at Torquay show that such sections are better than any other for the after body.

The trochoidal wave, as already observed, for its configuration is always dependent upon its relative length and height,† and travels with the speed of the ship it follows; whereas, the "wave of translation" is always of the same form, no matter what its height or length, and travels with a speed due to its own length and depth of water, as shown in the table, page 121.

We next come to consider Mr. Scott Russell's important discovery that a vessel, to avoid undue resistance from wave making must have a length of bow equal in length to the length of the wave, which travels at a speed equal to the speed proposed for the ship; and the length of the stern must be equal to two-thirds that of the bow, or of half length of a cycloidal

* *Vide* "Transactions of Institution of Naval Architects," 1881 (see farther on).

† A trochoid is thus formed. A semicircle is divided by aid of a protractor into any number of equal parts as shown by the lines 1, 2, 3, 4 (Fig. 69). The base line 0, 5 is also equally divided into the same number of parts, and the lines 1, 2, 3, 4, corresponding with the angular divisions of the semicircle, are projected at each division. Horizontal lines parallel to the base

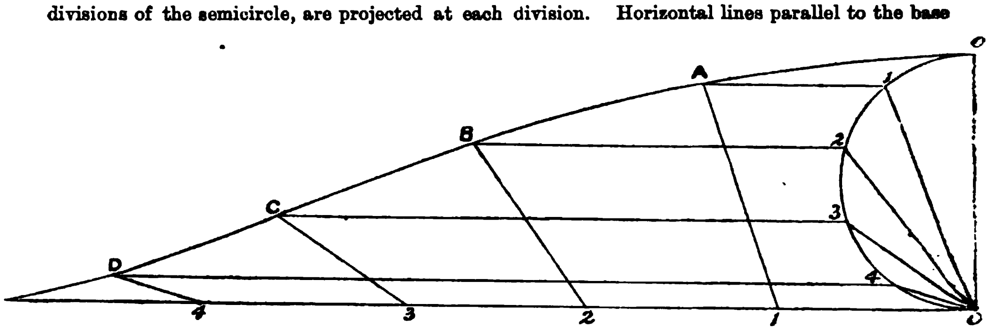


FIG. 69.

line are then drawn from 1 to A, from 2 to B, from 3 to C, and so on. Where the lines intersect at A, B, C, D, are points in the curve. The usual plan, instead of drawing the horizontal lines, is to take the heights (perpendicular to the base line) from the intersections with the circle and transfer them to A, B, C, &c.

wave which travels at the proposed speed, waves formed in deep water being now alone considered.* The ratio of fore body to after body, therefore, becomes 0·562 : 0·375 ; and, as the lengths of waves vary as the squares of their velocities, the requisite lengths of water-line for any given speed, V, proposed for a vessel will be found from

(F) Forebody = $\cdot 562 \times V^2$.

(A) Afterbody = $\cdot 375 \times V^2$.

The whole length of water-line for the given speed will therefore, according to this theory, be $F + A \times V^2 = \cdot 562 + \cdot 375 \times V^2 = \cdot 937 \times V^2$, or say the speed to be attained without undue resistance from wave making, is to be eight knots: the square of 8 is 64, and $64 \times \cdot 937 = 60$ft., the required length of water-line. Thus the speed will be approximately $\sqrt{F + A}$ before wave making tends to increase the growth of the resistance beyond the square of the speed in well-formed vessels. In other words, the speed attainable without undue wave making is nearly as the square root of the length; or the resistance until wave making becomes a pronounced feature is as the square of the speed. Of course it must be understood that this is not the limit of speed at which a boat 60ft. long can be driven; it simply indicates the speed which can be reached before the resistance due to wave making tends to make the general total resistance grow faster than the square of the speed. In practice, looking at the variety of conditions which have to be fulfilled, it has been observed that the speed a yacht can be driven

* The relative difference in the form and energy of waves created in deep or shallow water is here referred to, and not the general character of resistance due to depth of immersion. It is often supposed that a body moving in water meets resistance in proportion to the depth of its immersion. Mr. Froude, however, in his report on resistance, clearly shows that relative depth of immersion has little to do with the resistance of bodies moving as near the surface as the deeper part of a ship does. As a body is immersed more and more deeply below the surface water of greater density is displaced, and the frictional resistance varies as the density of the water; but, as a matter of practical usefulness, the increase of density for any depth likely to be reached by the immersed portion of a vessel is so small that it is safely neglected in all calculations of ship resistance. An opinion is also sometimes entertained that water is more difficult to displace or push on one side as the depth increases; this, however, is not the case. In reference to this subject, Sir W. H. White, in his work on "Naval Architecture," says: "If a plane were immersed very deeply, it would create little or no surface disturbance, and therefore require less force to propel it at a certain speed, than would a plane of equal immersed area moving at the surface with a portion situated above that surface. This statement is directly opposed to the opinion frequently entertained, which confuses the greater *hydrostatical* pressure on the plane, due to its deeper immersion, with the dynamical conditions incidental to motion. If the deeply immersed plane were at rest at any depth, the pressures on its front and back surfaces would clearly balance one another. When it is moved ahead at a uniform speed it has at each instant to impart a certain amount of motion to the water disturbed by its passage; but the momentum thus produced is not influenced by the hydrostatical pressures on the plane, corresponding to its depth of immersion. Water is practically incompressible. Apart from surface disturbance, the quantity of water, and therefore the weight, set in motion by the plane will be nearly constant for all depths, at any assigned speed. In other words, if there were no surface disturbance, the resistance at any speed would be independent of the depth."

rarely exceeds, in knots per hour, the square roots of their length multiplied by 1·25, or $\sqrt{L} \times 1.25 =$ speed.

No doubt there are several well-authenticated cases where a sailing yacht has been driven for a brief spell at a greater speed than the observed speed just quoted; and it is claimed for the Sappho, American yacht, that she, for a brief time, made a speed of 16 knots an hour during a run of 315 miles* in twenty-four hours in crossing the Atlantic in 1869, the average hourly run being 13·1 knots. The Sappho is 121ft. on the water line, and the speed of 16 knots would therefore equal $\sqrt{121} \times 1.455$. It should be noted that $\sqrt{121} \times 1{\cdot}25$ would give 13·75 knots.

The greatest speeds of sailing yachts observed by the author for a sufficient duration of time to be trustworthy, were as follows:

Length of L.W.L.	Speed for $\sqrt{\text{Length}}$	Actual observed Speed.	Length of L.W.L.	Speed for $\sqrt{\text{Length}}$	Actual observed Speed.
FT.	KNOTS.	KNOTS.	FT.	KNOTS.	KNOTS.
25·0	5·00	6·5	63·3	7·96	10·0
30·0	5·48	7·0	81·0	9·00	11·3
36·0	6·00	7·5	100·0	10·00	13·0
41·0	6·40	8·0	121·0	11·00	13·6
50·0	7·07	8·8	136·0	11·66	15·0

As already hinted a sailing yacht has to fulfil so many conditions, and the means of driving her are so limited, that the "actual observed speed" given above does not represent the maximum speed a similar yacht could be driven at if she had powerful engines in her, any more than $\sqrt{L} \times 1{\cdot}25$ represents the limit of speed a yacht can be driven at under sail. Steam launches we know can be driven at speeds double that which may be suitable, from an economical point of view, to their lengths, by putting the necessary power into them; and a man may propel a wager-boat far beyond the speed at which she first begins to sensibly form waves.

The case of the fast steam launches also shows that Mr. Scott Russell's theory as to the relative length of fore and after body does not hold good when high speeds have to be made; that is to say, speeds far in excess of the speeds appropriate to the total length. At such speeds it is found more advantageous to have the run at least as long as the bow, or longer by virtue of a raking midship section, for two principal reasons. Firstly, the bow wave-making resistance appears to increase in a uniform ratio (see farther on), but not so the other wave-making, which at high speeds assumes very formidable proportions in its growth. It has, in consequence,

* Miles and knots are used throughout this work as convertible terms.

been found advantageous, so far as torpedo boats are concerned, to have the stern somewhat longer than the theory prescribes, and consequently finer. These facts will in a way explain the cause of early experimentalists who towed small models at great speeds, arriving at the conclusion that the "cod's head and mackerel's tail" form is the "form of least resistance." A long run is also of great advantage for developing the efficiency of the propeller.

In the table which here follows, the total lengths of water-line (made up by bow and stern) for various speeds are given; and it must be distinctly understood that these are only intended to show the probable speed a well-formed vessel can be driven before the resistance due to wave-making begins to increase much faster than the square of the speed.

Knots.	Length of Forebody.	Length of Afterbody.	Total Length of Waterline.
	FT.	FT.	FT.
1	0·54	0·39	0·93
2	2·18	1·56	3·74
3	4·91	3·50	8·41
4	8.73	6·24	14·97
5	13·65	9·75	23·40
6	19·55	14·04	33·59
7	26·75	20·11	46·85
8	35·94	24·96	69·90
9	44·22	31·59	75·81
10	54·60	39·00	93·60
11	66·06	47·19	113·25
12	78·62	56·16	134·88
13	92·27	65·91	158·18
14	107·00	76·44	183·44
15	122·80	87·75	210·55
16	139·70	99·84	239·54
17	157·80	112·70	270·50
18	176·80	126·30	303·10
19	197·10	141·80	338·90
20	218·40	156·00	374·40

The Stream Line Theory of Wave-making.

It has already been pointed out that Mr. Scott Russell's theory as to the creation of waves by vessels moving in water is not accepted as the true one; but there is a general agreement that the stream line theory as elaborated by the late Professor Rankine is a correct solution of the primary cause of wave-making. The late Mr. William Froude, in elucidating this theory, first dealt with a perfectly submerged body placed at some distance below the surface of a "perfect fluid" moving towards the body.

In Fig. 70 let A be the parallel streams of this perfect fluid flowing towards the vessel B; then as the bow is approached a pressure is met and the breadth of these streams is increased, their direction altered in a variety of ways, as at C C, and the velocity of the streams is checked; as the sides of the vessel are reached the pressure diminishes and the breadths of the streams begin to narrow again as at D D, and their velocity is increased; at E the streams expand again, and once more lose velocity; and, finally, resume their normal condition and velocity some

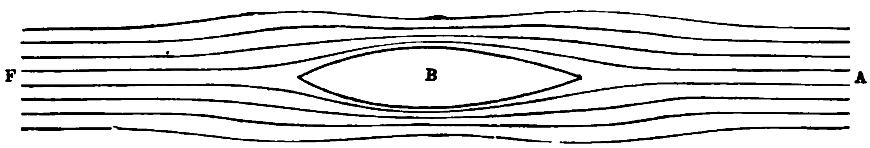


FIG. 70.

distance astern, as at F. By a series of convincing experiments Mr. Froude showed that all these pressures would exactly balance each other, and the fluid exerting them would not in consequence carry the submerged body along with it one way or the other; and by the same reasoning, if the body were moving instead of the fluid it would meet with no effective resistance, either from friction or pressure on the fore part, or any other part.*

It must be understood that this statement refers to a "perfect fluid" only, and a body moving beneath the surface of sea water is affected by

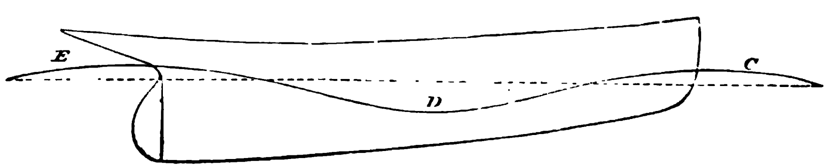


FIG. 71.

other conditions. Sea water, owing to its viscosity, exercises considerable frictional resistance to a body moving in it, as already explained; but beyond that resistance, a body submerged some distance below the surface, as a fish for instance, experiences no other resistance; in fact, any form, such as a cylinder with conic ends might be moved beneath the surface without making any apparent commotion, and the water would appear to be entirely undisturbed (see foot note, page 125).

The case, however, is entirely different when bodies partly submerged move along at the surface of the water. The same phenomena take place

* *See* "Proceedings of the British Association," Aug. 31, 1875.

with regard to the streams as when the body was submerged; but the immediate consequence of the retardation and acceleration of velocity which the streams undergo at the bow, middle, and stern are, that wave crests are formed at the points C C and E E, and wave troughs at D D (Figs. 70 and 71).

Thus, according to the stream line theory, the various pressures ought to equalise each other; and if force is expended in driving up a bow wave it ought to be restored by the pressure the water exerts in closing in around the stern. This condition is analogous to a statement that the cavity left in the water at the stern, as the vessel advances, ought to absorb the water pushed out of the way by the bow. This, no doubt, is a partly true state of the case, but no such reciprocal action can take place, as much of the bow wave will be spread out on the surface beyond the reach of the replacing water. Yet that the energy absorbed in creating a bow wave can be partly restored by waves made near the stern is well established, and the form of the hull, especially at the stern, plays an important part in this relation. It will be seen upon reference to Figs. 70 and 71 that the outlines of two waves are given (but not drawn to scale nor exact in portrait). C is the bow or divergent wave, and D the trough of a transverse wave with its crest under the quarter. A third or stern wave can be taken as partly diverging and partly transverse at about E.

The character and influence of these waves on resistance were thus described by the late Mr. W. Froude: "The inevitably widening form of the ship at her entrance throws off on each side a local, oblique wave of greater or less size according to the speed and obtuseness of the wedge, and these waves form themselves into a series of diverging crests (A A A, Fig. 72), such as we are all familiar with. These waves have peculiar properties. They retain their identical size for a very great distance with but little reduction in magnitude. But the main point is that they become at once disassociated from the vessel, and after becoming fully formed at the bow, they pass clear away into the distant water and produce no further effect on the vessel's resistance. But besides these diverging waves, there is produced by the motion of the vessel another notable series of waves which carry their crests transversely to her line of motion. These transverse waves, when carefully observed, prove to have the figure shown in detail (B B B in the diagram). In that diagram there is shown the figure of a model which has a long parallel middle body, accompanied by a series of these transverse waves as they appear at some one particular speed with the profile of the series defined against the side of the model. It is seen that the wave B is

largest when the crest first appears at the bow, and it reappears again and again as we proceed sternwards along the straight side of the model, but with successively reduced dimensions at each reappearance. That reduction arises thus: In proportion as each individual wave has been longer in existence, its outer end has spread itself further into the undisturbed water on either side, and as the total energies of the wave remains the same, the local energy is less and less, and the wave crest as viewed against the side of the ship is constantly diminishing. We see the wave crest is almost at right angles to the ship, but the outer end is slightly deflected sternward from the circumstance that when the wave is

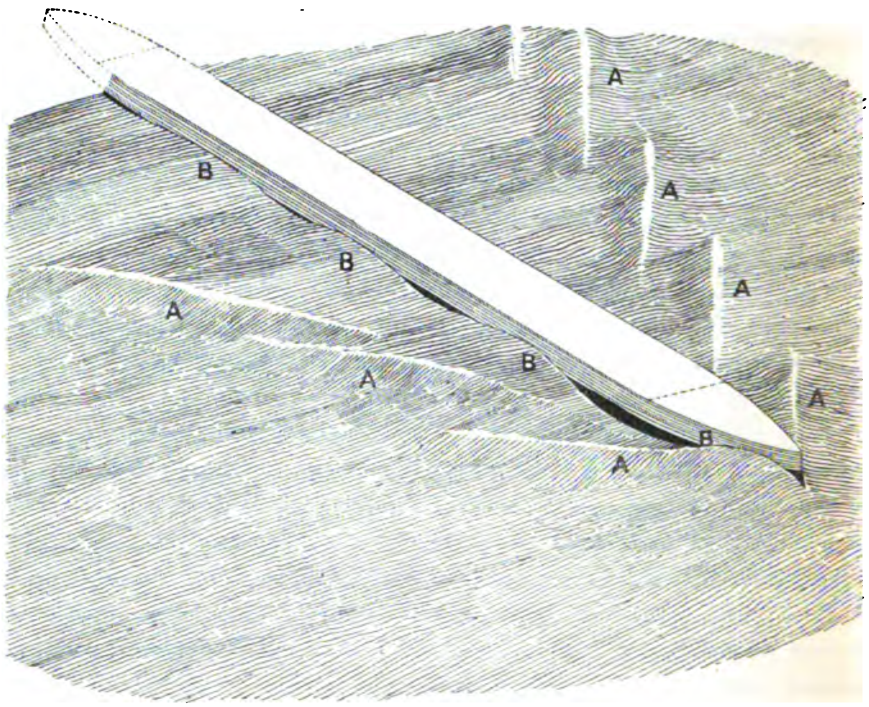


FIG. 72.

entering into undisturbed water its progress is a little retarded, and it has to deflect itself into an oblique position, so that its oblique progress shall enable it exactly to keep pace with the ship. The whole wave-making resistance is the resistance extended in generating—first, the diverging bow waves, which, as we have seen, cease to act on the ship when once they have rolled clear of the bow; secondly, those transverse waves, the crests of which remain in contact with the ship's side; and thirdly, the terminal wave, which appears independently at the stern of the ship. The latter wave arises from causes similar to those which create the bow wave, namely, the pressure of the streams, which, forced into divergence there, here con-

verge under the quarters, and re-establish an excess of pressure at their meeting."

It must not be supposed that a number of these transverse waves (B) will be found along the sides of vessels formed like sailing yachts, which have, as compared with steam ships, short and curved middle bodies; but one or more will exist, and be very pronounced at high speeds. They can, however, be seen distinctly enough along the sides of steam yachts which have a great length of straight middle body, or "parallel middle body," as Mr. Froude termed it. This feature will also be observable in sailing yachts which have this middle straight of breadth, but is more particularly apparent in the case of long steam yachts at speeds of 11 knots and upwards. The

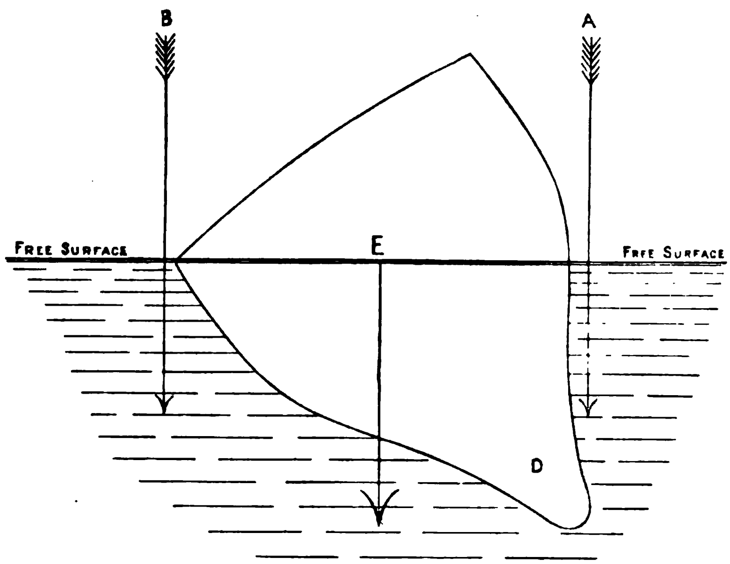


FIG. 73.

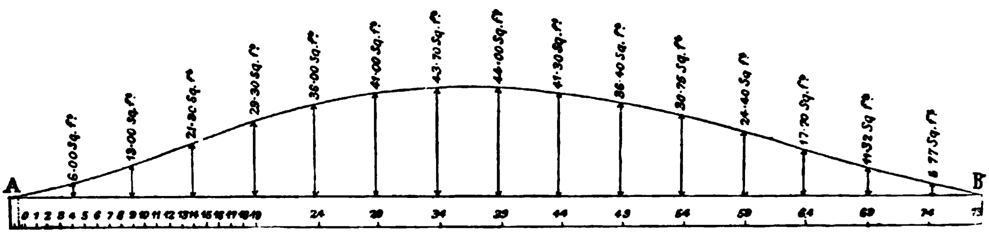
large transverse wave made by sailing yachts is generally most pronounced on the weather side if the yacht be of deep form and much inclined, although the case would be different with a shallow yacht. The explanation of this apparently strange feature is simple enough; the transverse waves, being long (relatively to the length of vessel and length of the diverging waves), have a natural tendency to be deep also, and therefore depth of action is congenial to their formation. If then, a vessel be heeled as shown by Fig. 73, the deep part of the displacement, D, will be plumb under the free surface of the water on the weather side (see the arrow A), and far from the free surface on the lee side (see the arrow B), thus causing the deeper transverse wave to appear on the weather side. At first sight it would seem that putting the deep bilge far

under the water on the lee side would have the effect of making a larger wave, but it will be seen that the bulk of the deep displacement becomes transferred to the weather side of the middle line (see the arrow E) of the plane of flotation. Of course, if the yacht retains an upright position, the transverse wave formed and along under the bilge will be the same on either side.

This is an important matter which will be again dealt with, but to assist in its elucidation it will be here necessary to introduce

THE WAVE FORM OF BODY THEORY.

In contradistinction to the theory that lines of any shape materially influence the resistance a vessel meets with in water, a variety of arguments have been from time to time put forward to prove that "lines" have little or nothing to do with the matter, providing the displacement is disposed of longitudinally in a certain rate of progression. Forms of all kinds of fish



CURVE OF AREAS OF SECTIONS.

FIG. 74.

have been analysed with a view of proving that the longitudinal disposition of a vessel's displacement should be similar to the disposition of the bulk or body of some swift fish, and the experiments made with submerged models seemingly indicated this view of the case to be correct. But it is now known by the light of the "stream line" theory that the resistances met with by submerged bodies and those floating at or near the surface are so different in cause and effect, that it is easy to understand the deceptive nature of former experiments carried on under water.

However, there is not much doubt that the ratio of the growth of the displacement in the fore body, and similarly the decrease in the after body, does influence resistance, and Professor W. Froude in all his experiments with models carefully recorded their "curves of displacement," or, in other words, curves showing the ratio of increase of area in the vertical sections, commencing with the stem or sternpost, and terminating with the greatest transverse section, commonly termed the "midship section." In Fig. 74 A—B represents the length of load water-line of a yacht, and the ordinates at the stations 79, 74, 69, &c., represent on a convenient scale

the area of each vertical transverse section at the stations. The curve is swept in through the ends of the ordinates.

This wave form theory has received a great deal of attention since it was elaborately put forward by Mr. Colin Archer, of Laurvig, Norway, in 1877. Mr. Archer appears to have adopted Mr. Scott Russell's theory, that a wave formed bow will produce its counterpart on the water, but contended that it is not *wave lines* which are required to do this, but a wave form in the longitudinal disposition of the displacement. Mr. Archer thus summed up his theory: "It is stated that the water which is excavated by a ship when she is moved through the fluid is carried away to a distance (not the identical particles of water in her track, but a corresponding mass), by a solitary carrier-wave, or wave of translation: and that the cavity formed by her largest section is filled up by a wave of the second order, or a common oscillating wave of the sea; therefore the entrance or fore body should correspond in length and form to the length and form of the carrier wave, travelling at the speed of the ship, and the run, or after body to the length and shape of the front of an oscillating wave. If so formed the ship will meet with a minimum of resistance in her progress."

To illustrate his theory, Mr. Archer took a rectangular block, and made all the water-lines (from the water surface to the underside of the model) correspond with the load water-line, which was a wave line. Each cross section was, of course, rectangular, of the same depths, but of different breadths. This model, as will be seen, not only had wave lines in her bow, but necessarily had a "wave form" of displacement for the bow (see A, Plate I.), which illustrates equally the curve of the water-lines and the curve of displacement. Mr. Archer next decreased the *depths* of all the sections, but kept their respective areas exactly the same. The result is shown in rectangular sections by B, and in semicircular sections by C (Plate I.), where the keel outline is curved to accord with the diminished depths, and the water lines become convex instead of waved. This model, Mr. Archer contends, has all the essential features of the wave entrance although there is not a wave line in it.

In narrow vessels the rounding up of the fore foot is almost a matter of necessity, as otherwise a heavy and injurious, not to say useless, quantity of dead wood would have to be carried in order to carry out the form; and even in broad vessels it is found advantageous to lighten the fore end as much as possible by cutting away the fore foot, or gripe, as it was formerly termed.

It should also be remarked, in reference to the convex water-lines, that they existed in many narrow vessels by the best designers, and also in all the broader vessels of the most recent type, which have

small depth of section in the fore body. For the entrance "round lines" seem an advantage from a frictional point of view, although their effect, of course, is to increase the angle of entrance. It does not follow that because hollow lines are of no advantage at and near the load water-line in an entrance which has very little depth of vertical section in the fore end of the bow, that there might not be some advantage to be derived from them in a vessel whose entrance is deeper.

Round or convex water lines are by no means new features in vessels; in fact, they are to be found in the oldest models of which we have any record, just as hollow lines are; but it is essential to remember that where there is a great depth of vertical section a sort of $\supset$ shaped bow will be the result if convex lines are adopted in the bow, which will push a huge wall of water in front of it, and much increase the head resistance. (See "Resistance" further on.) The convex bows of the modern shallow-bodied yachts also push off a considerable wave from each side of the bow, but the resistance due to these waves is not so great as that due to the transverse waves created amidships in deep-hulled vessels with comparatively sharp or hollow wave-line entrances. The transverse wave making it need scarcely be said is much less in shallow-hulled yachts, which with convex bow lines turn off large bow waves; but it would be a difficult matter to measure the resistance due to each, although the total resistance could be easily measured by experiment.

Mr Archer thus propounded his theory as to water lines:

"An advocate of wave water lines might object that, since the areas of the vertical sections are determined by the form of the water lines—these lines determining the lengths of the ordinates by which the vertical areas are computed—and since a certain progression of these areas is essential, therefore a certain curve for the water lines is equally essential. In this case the required progression of area has no doubt been brought about by using water-lines of a certain geometrical curve; but the same result may be brought about in a great variety of ways. It is important to remember that the *shape* of the vertical sections is, within reasonable bounds, immaterial as far as the wave theory is concerned. The same liberty, claimed in this respect for the midship section, applies with equal justice to any other section. That which is the material point is, that each cross section shall have the exact area which its station in the ship assigns to it, since, if this is the case, the water will be displaced precisely in the required progression. We are, therefore, at liberty to alter the shape of the sections of our fore-body at pleasure. (It is always understood that due care is taken to ensure a 'fair' body, and to avoid abrupt curves.) We may, for instance, round up the under part of our block-model—that

part which in a vessel is called the 'forefoot'—as shown in B, always provided the same area which is taken away from the lower part of each section is added to it at another part, namely, at the side. The water-lines would then take the shapes shown in the half breadth plan C; and we see they have now lost all resemblance to a wave-line. And yet we have not in the slightest degree deprived our entrance of its wave character.

"The model C has the same length, the same area of midship section, as well as every other vertical cross section, and consequently the same volume of displacement as A or B. But instead of beginning with water-lines, or something representing them, it is constructed entirely by vertical sections, altogether disregarding what shape the water-lines may chance to take. Instead of rectangular sections we adopt semi-circular ones. Knowing what the areas of the sections at the various stations are, either by calculation or by measurement from the first block, we have for the length of radius for any section, radius $= \sqrt{\frac{2 \text{ area}}{\pi}}$.

"Having drawn the various sections in the body plan by simply striking off the half circles (or the quarter circles for one side), we may then proceed to fill in the water-lines; and we find that, so far from any one of them being

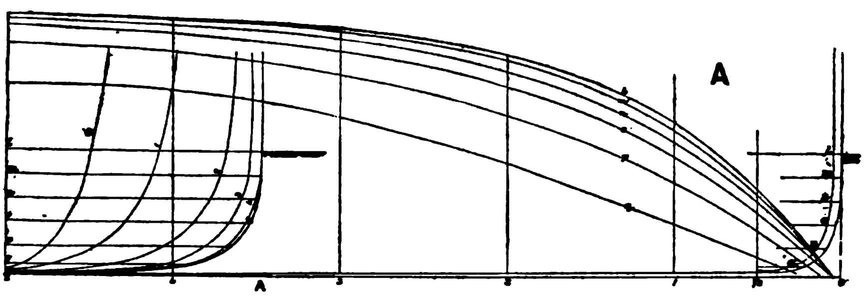


FIG. 75.

a curve of versed sines, there is not a hollow to be found at any part of them. And yet we hold that it is a 'pure wave' entrance. It evidently displaces the water in such a progression as exactly to give off the necessary quantity for supplying the swell of the wave; in the same progression, in fact, as the first block model, and as Scott Russell's 'wave water-line' model."

The late Mr. W. Froude in 1876-1877 was engaged in making some experiments for the Admiralty, and an analysis of the results given in his report throws some light on the influence form of body, or the disposition of displacement, has on resistance. Of the many forms tried one was that of a gunboat of the ordinary dockyard type, the bow of which is shown by A, Fig. 75.

The speed required was nine to ten knots, and for various reasons it was desirable to obtain this speed as economically as possible. It would take up

too much space to analyse all the various forms Mr. Froude experimented with, but the main results were distinctly exemplified by four of the models.

Upon reference to Fig. 76 it will be seen at first sight that one of the models, B, had a bow very similar in form to that of A, Fig. 75; but such was by no means the case. The fore foot, it will be seen, was very considerably rounded up, and the water-lines terminate farther back in consequence, and, owing to the increase in the breadth, and consequent increase in the area of midship section, the displacement became concentrated more amidships, with the result that longer ends were obtained by practically doing away with the length of middle body observable in A, Fig. 75. If the stem of B had been like A's, and the various half breadths on No. 1 section maintained, the fore ends of the water-lines would have shown some hollow or contrary flexure in being drawn out to the stem; the displacement, however, at the fore end would have to be somewhat added to, and the displacement curve would not have shown such a fine end. In

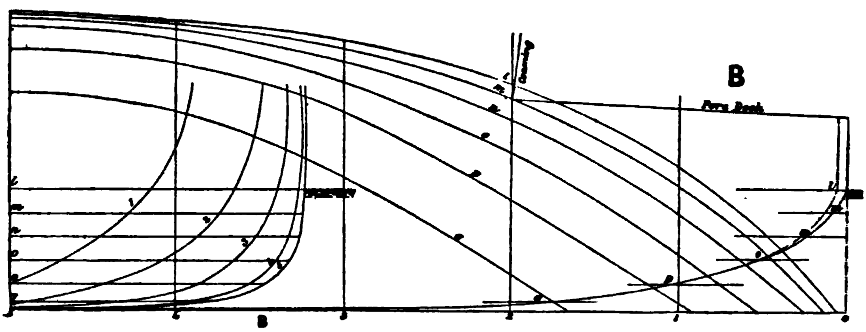


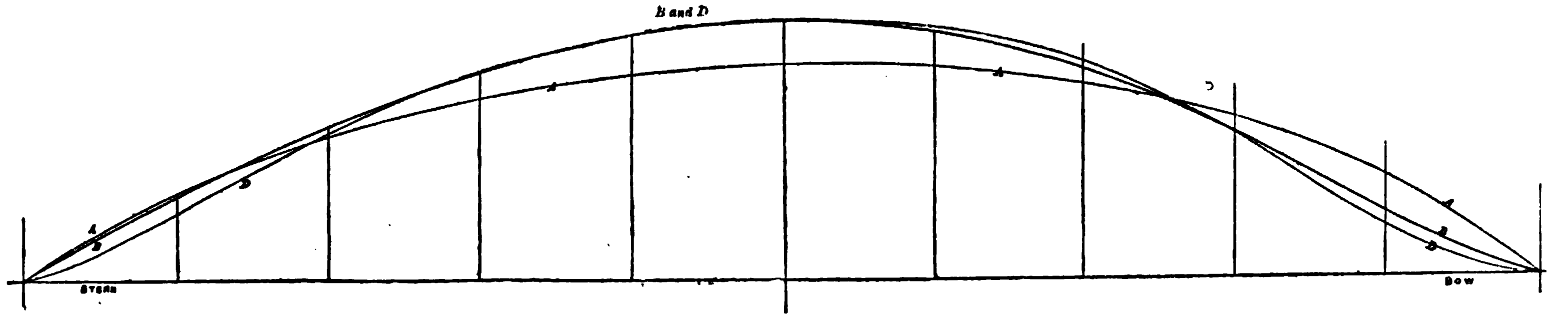
FIG. 76.

Fig. 77 the displacement curves of A, B, and C, are shown, also the lines of the model C, and it will be seen that a "wave form" of displacement can exist even with round full lines such as those of B.

The displacement curve of C is practically the same as that of D, Fig. 78, but it will be observed that the lines of the fore body differ somewhat, and C has a fore foot like A's, whilst D's is rounded up like B's.

The after body of B, C, and D were practically the same; but that of D was a trifle the finer. The dimensions and weight of these models (given in terms of weight or displacement of those for the ship) were as follows:

	Length.	Breadth.	Draught.	Total Displacement
	FT. IN.	FT. IN.	FT. IN.	TONS.
A	85 0	26 0¼	6 3	254
B	"	30 1	"	265
C	"	"	"	258
D	"	"	"	256
D_1	"	"	"	256



Curves of Displacement

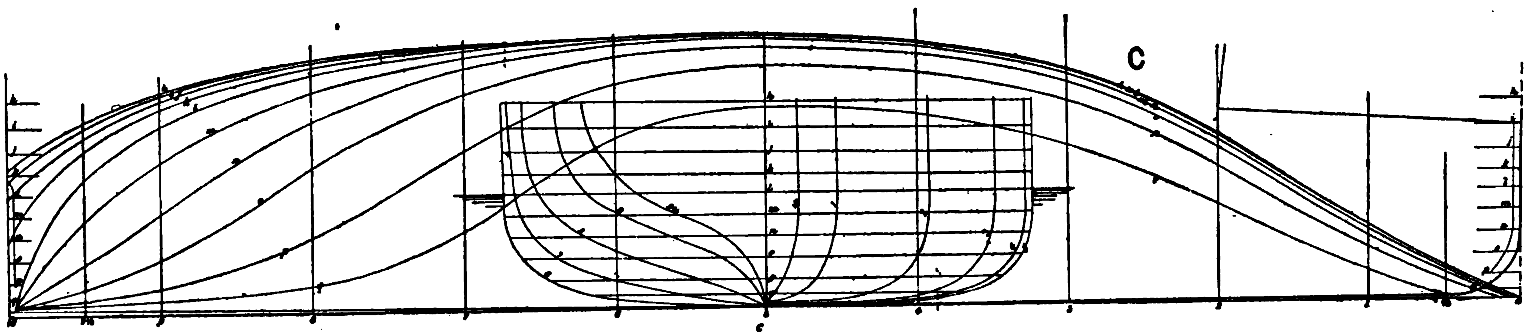


Fig. 77.

The smaller displacement of A is attributable to her smaller midship section; and the excess displacement in B, as compared with C and D, is attributable to her fuller bow. The resistances of these models (in terms for the ship) were as follows:

	5 Knots.	6 Knots.	7 Knots.	8 Knots.	8½ Knots.	9 Knots.	9½ Knots.
	Tons.	Tons.	Tons.	Tons.	Tons.	Tons.	Tons.
A	0·28	0·44	0·69	1·13	1·46	1·98	2·78
B	0·29	0·45	0·67	1·05	1·31	1·63	2·11
C	0·29	0·44	0·62	0·89	1·11	1·40	1·90
D	0·27	0·42	0·64	0·87	1·06	1·29	1·60
D_1	0·27	0·44	0·64	0·89	1·08	1·28	1·57
E	0·27	0·40	0·53	0·75	0·85	0·98	1·15

D^1 is D with a slightly more flaring bow, as shown by the dotted lines in Fig. 78.

E is a vessel with similar sections to D, but reduced in area and spaced out so as to make the length on water-line 125ft., and the displacement still 256 tons. The relative resistances are also shown by curves on Fig. 79.

Of course it must be understood that this was a case where for a fixed displacement the length of load line was limited, and the draught of

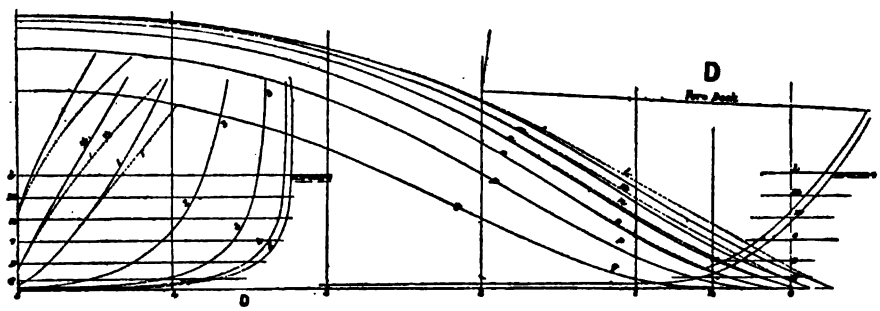


Fig. 78.

water also; and, as a matter of course, if the length could have been increased as it was in the case of the model E, the resistance would have been reduced to a much greater extent than it was by increasing the beam. This is well illustrated by the curve for E. However, with the restricted means at disposal, by adding to the beam, and thereby increasing the area of midship section, a finer actual entrance and run were obtained with a continuous curve from end to end, the result being a large decrease in the resistance, or in the power required to attain a given speed. The experiment was, moreover, valuable, as it conclusively proved the fallacy of the assumption that the area of the greatest transverse section (midship section) can be a measure of resistance except for vessels of exactly similar form and proportions.

The results of the experiments just described with the A and B models, and many other similar experiments made by Mr. W. Froude, afford proof that a bow whose curve (page 132) of sectional areas, or displacement,

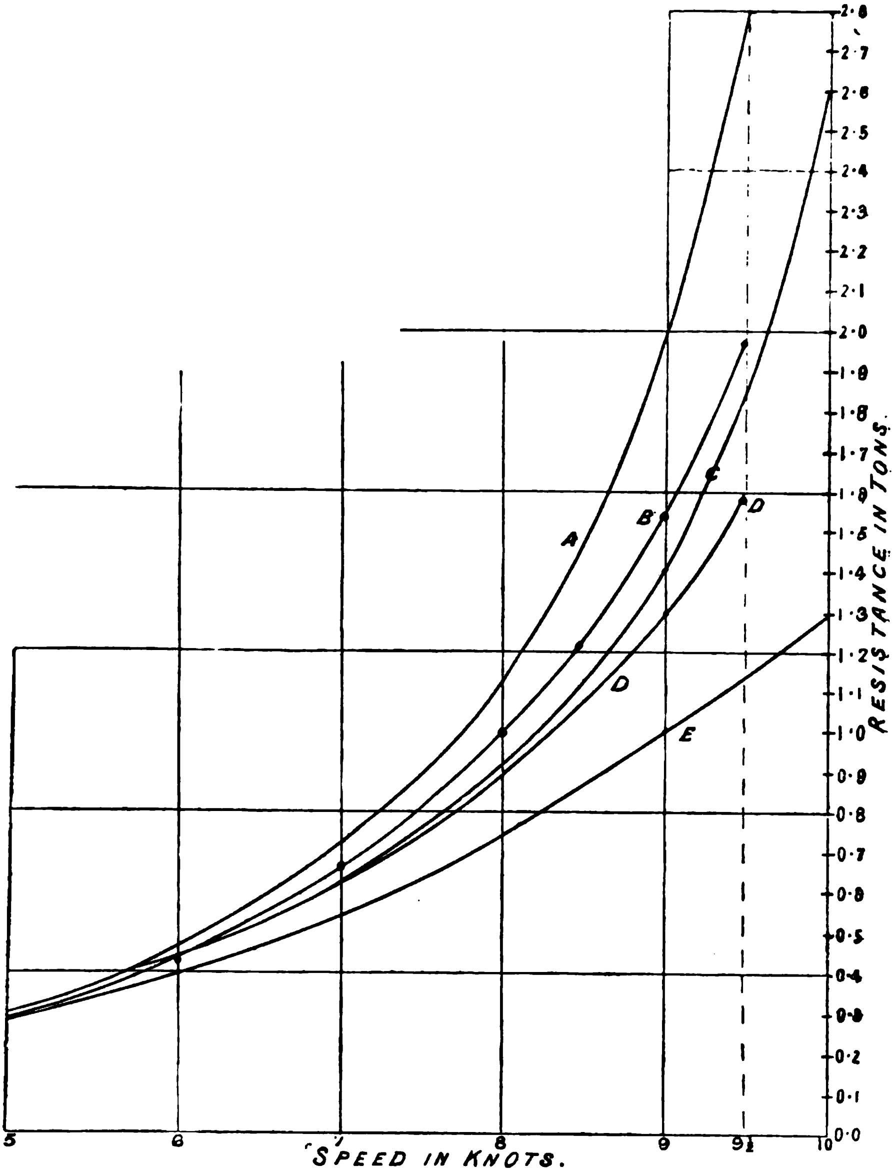


FIG. 79.

forms a curve of versed sines is of great value where a high rate of speed for any given propelling power is desired. The further experiments with C and D would also seem to indicate that there is, under certain conditions, some value in wave lines, although, as will be shown farther on, it does not

seem to be of much importance that the extreme fine ends of the lines should be retained, judging by the effect of cutting off those ends in D, and thus rounding up the fore foot.

Illustrated by the vertical sections it would appear that the U shaped vertical cross sections in the bow, like the model C has, have some advantage over V sections, like those of B model, but that V sections, like those in D model, with wave ends to the water-lines, are of the most value within the limits of the speeds experimented with.

There is, however, one important circumstance to consider in connection with this matter. The bow with U sections, as might be supposed, exhibited a much greater tendency to "bore" than those with V sections (see Fig. 80); and the fuller the water-lines accompanied by U vertical sections the greater the tendency, and this is a matter of some consequence,

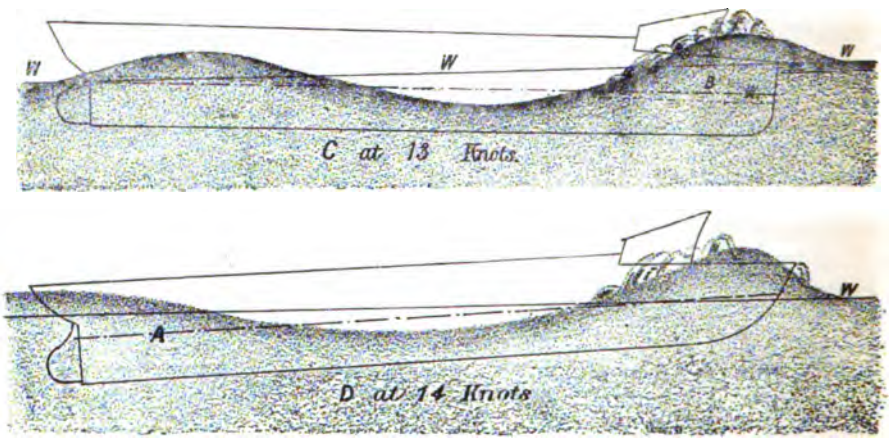


Fig. 80.

as "boring" will cause a very serious diminution of speed. Upon reference to the table of resistances it will be seen that D shows less resistance than D_1 at 8 and $8\frac{1}{2}$ knots, but at 9 and $9\frac{1}{2}$ when "boring" began to manifest itself, the more flaring bow, shown by the dotted lines of D_1 gave a smaller resistance.

In dealing with the question of a V form of vertical section for the bow as against a U form, it would be necessary to consider whether the vessel had to be driven at a speed when boring would be likely to occur. In the long narrow racing yachts, with very sharp V sections in the bow, boring is hardly likely to manifest itself to any serious extent; but in the shorter Itchen boats the case is different, the tendency being also magnified by the pressure of the enormous sails they carry. In such yachts, therefore, the flaring bow, or broader V section, such as their greater beam involves, is an advantage as against the bow with flatter sides, although

it is possible that there may be some loss of speed under certain wind forces when "boring" has not commenced to a serious extent.

As already described, the wave form theory has been no less busy with the after body than with the fore body, and the conclusion arrived at from the results of observations on actual vessels and experiments with models is, that the displacement of the after body should be about equal to that of the fore body, and be disposed of in a wave form. The experiments with the broad models made by Mr. Froude seemed to indicate that a clean flat buttock line at the quarter breadth as against a steeper one was an advantage; and also that fining the extreme after end so that the displacement took the form shown in the displacement curve (D, Fig. 77) was an advantage.

Mr. Colin Archer thus summarises his view of adapting the "wave form" theory to the after body: "The form of the after body is required by the wave theory to be such that an oscillating wave following a vessel with her own velocity shall at every point just fill the vacancy left by each succeeding section, and no more. This is an essential condition. Suppose, for instance, that at a given distance from the initial point of the wave—which will be at the largest section of the ship—a section of the said wave has attained to one-fourth the area of its greatest section. It is then required that room shall have been provided for sufficient water to form this area of cross section. A little farther aft the sectional area of the wave will have reached one-third of its greatest area; and room must by the time this is reached have been provided for this addition to the area of the wave section. The area of any vertical section in the after-body must therefore be the area of the midship section less the area which the swell of the wave supplies at that stage of its formation. The wave is not perfectly formed until the stern end of water line is reached. Therefore, what is wanting in volume at any point must be made up by the body of the vessel at that point. As the water flows to form the following wave from every point from which it has free access, we need not have the curve all one way."

Mr. Archer derived a radius for the trochoidal curve of sections for the after body from the following formula:

$$\frac{4\,D}{\pi\,M} - \frac{2\,L}{\pi}$$

Where D the displacement of the after body, L length of load line of the after body, M area of midship section, and π = ratio of diameter to circumference = 3·14.

It has already been stated that a valuable result of the long course of experiments carried on at Torquay for the Admiralty, has been to prove that the wave form of bow is of importance, if a high rate of speed is

sought for any given displacement, length, and propulsive power. Another valuable result was to confirm Mr. Scott Russell's wave length theory, that for ordinary speeds the after body can be shorter than the fore body, to the extent indicated by the theory (see page 125). This relatively shorter after body can also have greater fulness than the fore body, and the curve of areas of sections aft might form a trochoid; but so far as yet discovered a curve of versed sines, compressed into the relatively shorter length of after body would do equally well. If a trochoid is adopted Mr. Colin Archer's formula for determining the radius of the circle could be used if the displacement of the after body is known; or a simpler method would be to take $\frac{1}{10}$ the length of the after body and a function of half the area of the midship section, thus: $\frac{\sqrt{(M \times 0{\cdot}5)}}{2}$. The square root of the half area of the midship section is taken as approximately the diameter of a circle of that area, and half the square root will be the radius of the circle. The formula for the construction circle of the after body will, therefore, be

$$\text{Radius} = \frac{\frac{L}{10} + \frac{\sqrt{(M \times 0{\cdot}5)}}{2}}{2} = \frac{L}{20} + \frac{\sqrt{(M \times 0{\cdot}5}}{4}.$$

For projecting a curve of versed sines, or a trochoid, about 5 per cent. is added to the end of the load line forward and aft, in order to afterwards cut off the very thin termination of the curves.

Variations in the Ratio of the Growth of the Resistance at Progressive Speeds.

It was for a long time surmised that when once the wave-making period commenced the resistance would go on increasing as the square or the cube of the speed; but the ratio of the increase of the resistance may vary considerably, undergoing ultimate augmentation and diminution, after the wave-making speed has been reached. This feature of resistance in water it is true had long been known or suspected, but it was first brought to prominent notice by the performance of the fast steam launches built by Mr. Thornycroft. One of the first of these was the Miranda, 45ft. 6in. on the water-line, 6ft. 6in. beam, and 1ft. 9in. draught of water, displacement 3·72 tons. Sir F. Bramwell experimented with this launch, and, according to his report, the speed and corresponding resistance in I.H.P. were shown in Fig. 81. It will be seen that between five and seven knots the resistance grows very slowly, or at a less rate than the square of the speed. Between seven and eleven knots the resistance increased as the fourth power of the speed, and the curve shows some indications of a "hump;" after this hump has been passed over, between twelve and

fourteen knots, the increase of resistance had fallen again to nearly the square of the speed, but began to rise again after passing fourteen knots.

A want of knowledge of this variation in the ratio of the growth of the wave-making resistance no doubt led early experimentalists into many errors, which were further emphasised by neglect of the law of comparison for the speed of a ship to that of her model, the models usually having been towed at a speed equal to the assumed speed of the ship she represented. Mr. Froude conducted a number of experiments to ascertain how the wave system operates on a ship's resistance at different speeds, and he demonstrated that the transverse waves are the main cause of the variations in the growth of the resistance, and the author must here acknowledge his in-

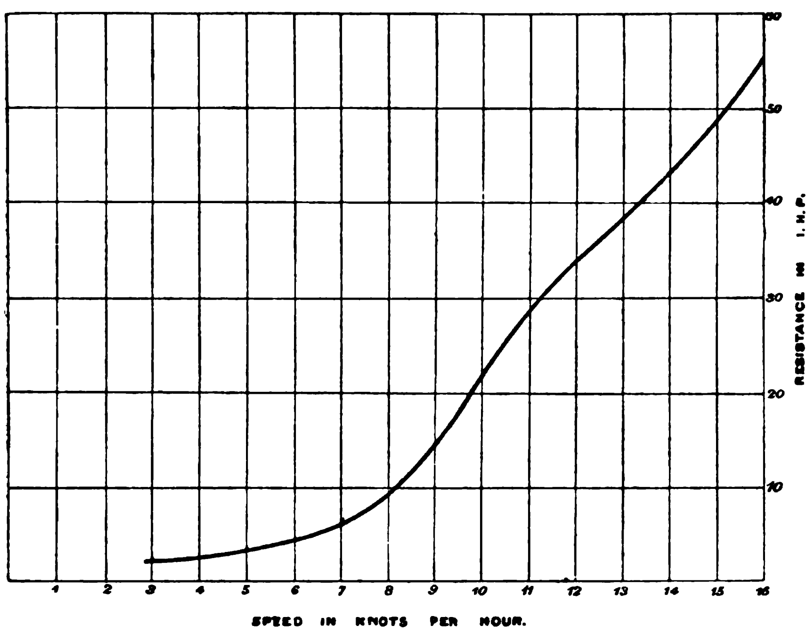


FIG. 81.

debtedness to Mr. R. E. Froude for a clear elucidation of the phenomenon. In the first place, it will be necessary to separate the resistance due to the diverging waves (D) formed by the bow from the resistance due to the transverse waves (T) formed by the bulk of the body. Also to disregard resistance due to friction which has a constant ratio of growth.

In Fig. 82, the curve D represents the resistance due to the bow wave (not to any scale). It will be seen that its ratio of growth is represented as practically constant, and its magnitude at any given speed will depend mainly upon the length and angle of the entrance.

It has already been shown that the speed of waves is governed by their lengths, and that the longer the wave the greater the speed; it

therefore follows that if the entrance of a ship tends to make comparatively long waves she can be made to travel at a speed nearly equal to the speed naturally appropriate to the wave before she commences to meet with undue resistance; and, further, it is an established principle that the longer the bow of a ship is made the longer will be the waves it will tend to create. Thus, for instance, supposing we take the curve of areas of sections for a yacht (see Fig. 74), and increase the spacing between the sections so as to bring her total length of water-line up to 100ft., it is obvious that the bow will be longer, and "finer," because it will make a smaller angle with the centre fore and aft line. The tendency of this bow will therefore be to make longer waves, and consequently the yacht could be driven at a proportionately higher speed before she met with undue

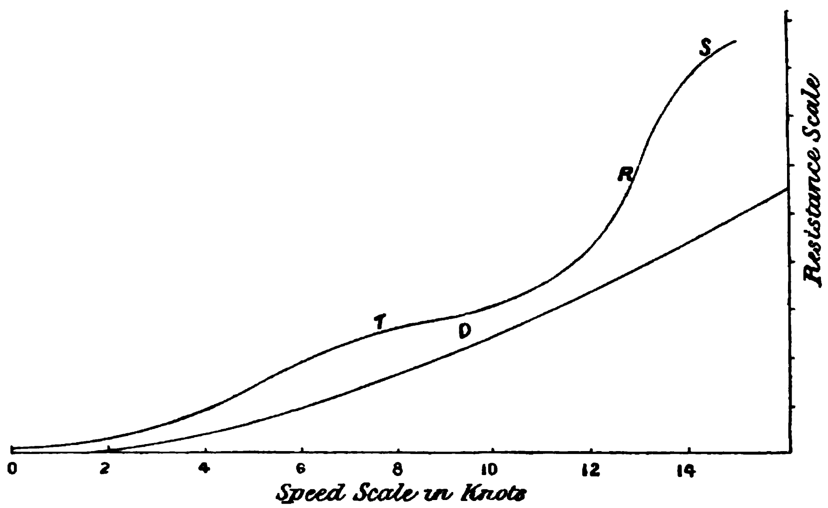


Fig. 82.

resistance from bow wave making. On the other hand, if she were shortened, still retaining the same sectional areas, she would have her entrance shortened and altered to a coarser angle; the waves the bow would tend to make would be shortened also, and have a less natural speed; consequently the yacht's speed would approach the speed of the waves earlier, and to get her beyond the speed of the waves would require an undue expenditure of power.

Thus the scale of the D curve, Fig. 82, depends mainly on the form of the fore part of the load line and lines near under it, because the diverging waves being comparatively short in the line of their motion, are consequently shallow, and are not affected by displacement at a low level. The D resistance will thus mainly depend upon the relative sharpness or bluntness of the bow. A long straight wedge-shaped bow will make similar

diverging waves at all moderate speeds, the scale of the waves being proportional to the scale of the square of the speed.

If the bow is hollow the wave making at low speeds will be dependent upon the sharp end of the bow, and the D resistance will be less at low speeds than with the straight bow before referred to; but this advantage need not exist at higher speeds, as then a vessel with a hollow bow would be turning off larger waves from her shoulder on each side, especially if much heeled, and she would then make larger diverging waves than one which has straight lines. Shortening back the lower water-lines (see D, page 136-7), and making the load water-line and those near it fuller, so as to bring about sections in the bow more V than U, tend to increase the D resistance and decrease the T resistance because the effective depth of the body is diminished. Whether or not there will be a net gain depends upon which resistance is the most potent at any given speed.

The character of the curve of resistance (T, in Fig. 82), due to the formation of transverse waves, is mainly governed by the form given to the displacement as depicted by the curve of areas, Fig. 74, page 132. It has already been pointed out that the transverse waves can be sustained with great depth of trough, and that deep displacement tends to create deep transverse waves.

A very broad keel, such as modern racing yachts have, will therefore tend to emphasise the formation of transverse wave-making resistance, although the increase to the resistance from this cause, so far as practice has yet gone, would be more than compensated for by the extra stability due to the lower position of the centre of gravity of the lead keel consequent upon its broader form. The same remarks apply generally if the stability is added to by deepening the keel indefinitely, although, of course, the thicker the keel the more it would tell in the resistance; and the smaller would be the gain in speed due to the extra sail-carrying power. This always assumes that the fore edges of the keel are pared away to make as fine and fair an entrance as possible. If the keel were thin, say from 1 to 4 inches thick, as it is in a fin keel, nearly the whole of the resistance, due to deepening the keel, would result from skin friction. The additional resistance, due to the extra depth being frictional, could be met, within the limits of stability, by proportionally adding to the sail area. This to a great extent explains why it has been possible to take so much advantage of under water depth in small yachts.

U sections in the bow and stern also tend to increase the depth of transverse waves, and therefore of the T scale of resistance, for the reason that they increase the effective average depth of body.

It therefore follows that with any given area of cross sections

deepening and narrowing them, or deepening the bilge, will increase the resistance due to tranverse wave-making; and with any given displacement, decreasing the areas of cross sections and increasing the spacing between the sections, so as to increase the length and decrease the angle of entrance (see the curve E, page 139), will tend to decrease the resistance due to transverse wave-making. Attempts, however, to measure the resistance from the angle of the bow would necessarily end in failure, as the general resistance rests on so many distracting conditions.

Deepening the bilge, although it would increase the resistance due to transverse wave-making, might considerably add to stability (see page 43).

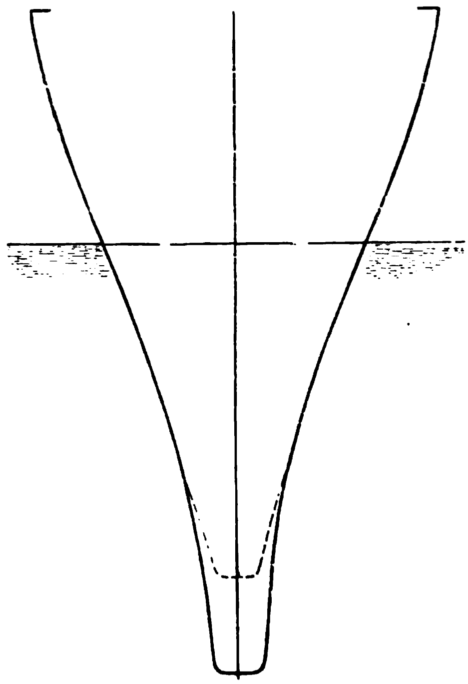


FIG. 83.

Making the cross sections in the bow, or stern, of the V form for *any given displacement* tends to add to the D resistance, because by spreading out the upper part of the sections the angle of resistance must be greater, and necessarily the resistance from the diverging waves will be greater. At the stern end this may be a distinct advantage; so far as the stern divergent waves are concerned, they are of less serious import in the general resistance than the tendency the stern might have to increase the transverse wave-making; and even at the bow there may be an advantage derivable from the V bow, as indicated on page 139.

This must not be understood to mean that if a vessel has U cross

sections in the bow, and they are altered to V sections, without altering the shape or increasing the angle of the upper water lines, that there would be an increase in the resistance due to the diverging waves; on the contrary, there would be a palpable decrease, as the displacement would be reduced, and so would the surface, and the fore end of the curve of areas would be finer. This will explain the cause of the advantages which have

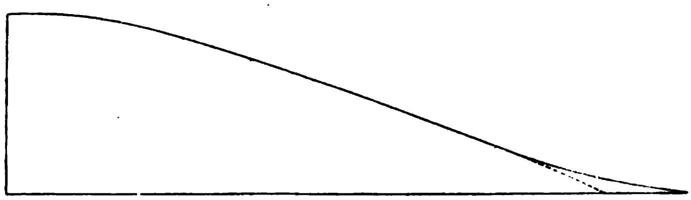


FIG. 84.

arisen by cutting away the fore foot of modern yachts, as shown by dotted lines in Fig. 83.

Alteration of character of curve of areas (see page 132), so far as it partakes of the nature of cutting off a small piece at the end of the curve of versed sines, does not much modify the curve of T resistance (see the dotted line Fig. 84), although it would necessarily add to the resistance (D) due to the diverging of bow waves. If the alterations referred to are made so as not to alter the total length of forebody (either by contracting the curve versed to *a b*, and then putting in a quasi-parallel body *b c*, or by stretching out the curve of sines to *a c*, as shown in Fig. 85, and then cutting off the thin piece), the result will be a combination of the effects of contracting the length scale and departing from the character of the curve of sines.

It may be inferred from what has gone before that, for any given

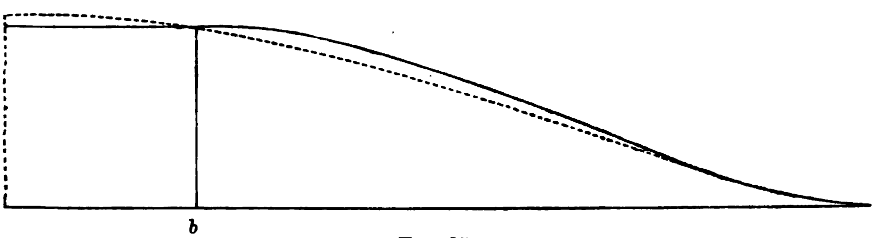


FIG. 85.

displacement and length of water-line, a vessel with a length of five times her beam and great comparative depth, would meet with less resistance than one of greater beam and less depth. This, however, is only a partly true state of the case. It can be assumed that two yachts are each to be 90ft. on the water-line, and each to have 150 tons displacement; but whilst one has a beam of 18ft. and a depth of under water body of 9ft., the

other has a beam of 25ft. and a draught of 6·5 only. These sections (see Fig. 86) can be taken as fairly representing yachts of the modern type and the type in existence up to the year 1888. At low speeds, up to three knots, when the resistance would be mainly accounted for by skin friction, the narrower vessel would get an advantage, provided they had equal sail area per square foot of immersed surface. If such an equality did not exist, the yacht with the larger proportion of sail spread to wetted surface would make the greater headway even at three knot speeds. This, of course, assumes that the quality of the sails and the quality of the surface for friction are equal. If the shallower hull has deep fixed keel or fin of the modern pattern and an immersed

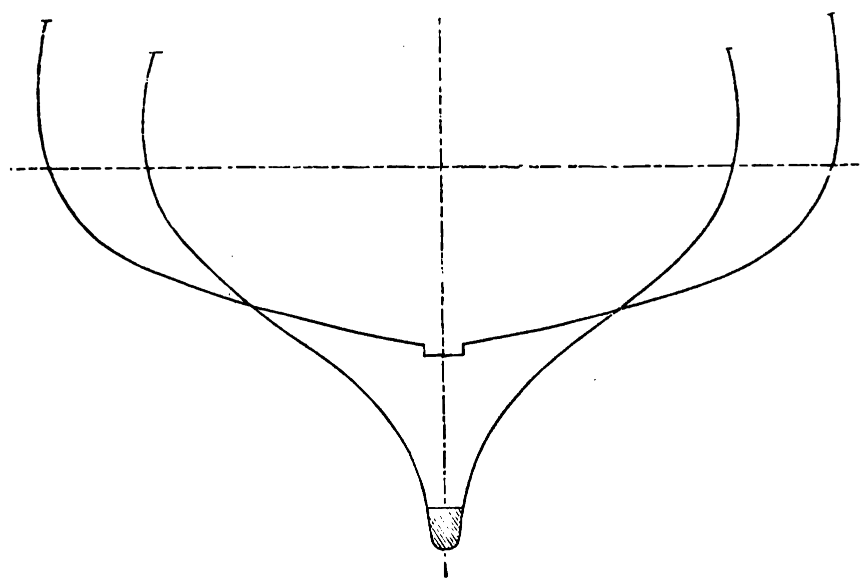


FIG. 86.

surface and sail spread equal to those of the narrow yacht, she would obtain an advantage when they arrived at the first stage of wave making, say at five knots, if the entrance of the broader boat should be hollow, as only the fine or thin end of the entrance would, as yet, be concerned in making the diverging waves at the bow, whilst the transverse wave-making would still be unimportant in either (see page 144). At higher speeds, however, say at seven or eight knots, when the broader vessel began to get her bow, as it were, into the waves, she would turn off diverging waves of greater size and at larger angles ; the narrower yacht would then get a considerable advantage, always assuming she had an equal sail spread per square foot of immersed surface, and per unit of displacement[1], and carried her canvas as effectively as the broader yacht.

This advantage would be maintained until the transverse wave-making began to form a larger part of the total resistance than the part due to the bow diverging waves, say at about eleven knots for lengths of 90ft. Then the shallower vessel would have the advantage, as her transverse wave-making would be relatively small. It must be recollected that the resistance due to the bow waves increases in practically a constant ratio, as already explained, whilst the ratio in the growth of the resistance due to transverse waves may rapidly increase.

It must be understood that the foregoing refers entirely to performance in smooth water, as performance amongst waves is influenced by many other qualities.

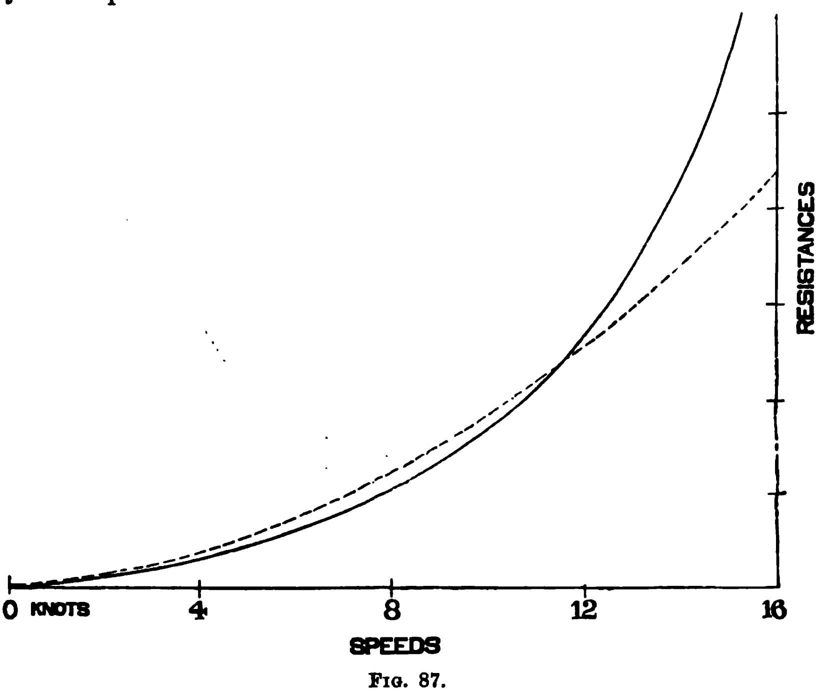


Fig. 87.

The character of the curves of the resistance of the two types is depicted in the diagram, Fig. 87, and exhibits the variation in the resistance of the broad and shallow, and deep and narrow type of vessel for any *given length and displacement.* The dotted line represents the curve of shallow type, and shows that the resistance is only increasing moderately at about eleven knots, whilst that of the deep model is rising very rapidly, and crosses above the curve of the other at about the speed named. The "speed scale" is made to suit a length of about 90ft.

We now come to consider the cause of the "hump" in the T curve of resistance as shown in an exaggerated manner at T, Fig. 88, page 150,

also in the more exact curve representing the resistance of the Miranda (Fig. 81, page 143).

The results of the experiments conducted by Mr. Froude pointed to the significant fact that the "hump," or, in other words, the sudden growth in the resistance when some particular speed was reached, was less pronounced when the curve of displacement approached a curve of versed sines or actually formed such a curve; but all trace of the hump was never obliterated if the speeds experimented with were sufficiently close together to ensure its not being passed over. This is a matter of some importance, because the "hump" rarely appears in the ordinary diagrams of speed curves; but this is only for the reason that the speeds experimented with

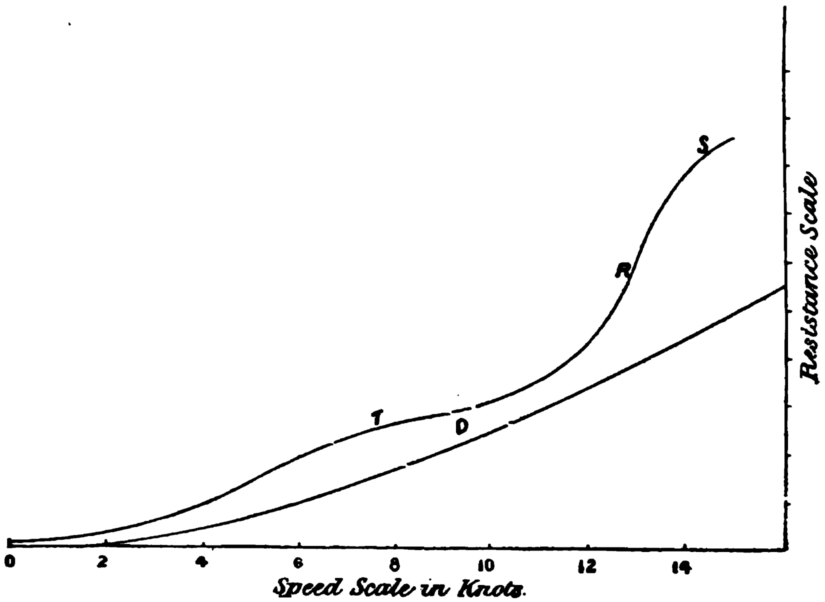


FIG. 88.

are few and far between; and, moreover, the effective horse power may have variations with the calculated or indicated horse power, which variations do not, of course, come out in ordinary trial trips on the measured mile. In the extended trials of torpedo boats and torpedo catchers made by the Admiralty, the "humpiness" of the speed curve is, however, apparent enough.

The humps which in the curve indicate a sudden increase in the ratio of the growth of the resistance do not appear, as will be readily understood, until the vessel arrives at a pronounced wave-making speed, which, as already explained, will be when the rate of speed per hour approximately equals the square root of the length of the vessel for fair forms.

Thus, in a 100ft. vessel of fair form it would begin to appear at about ten knots, and Mr. R. E. Froude, in some experiments he made with a model of a narrow and deep 3-tonner, found it apparent between 4¾ and 5¼ knots, which are about the speeds at which it might be looked for if ten knots is the speed for its appearance in a 100ft. vessel. After the "hump" is once surmounted the ratio of the growth of the resistance would fall again, forming a sort of flat place or hollow in the curve; but the growth in the resistance is subsequently so enormous that only torpedo boats have yet climbed the steep part of the curve R (Fig. 88) and got into the flat place again at S. It will be gathered that the steep part R occurs at different speeds according to the length of vessel concerned.

Suitably to what has been advanced, the first indication of the "hump" (T, Fig. 88), or great increase in the growth of the resistance, might be looked for in varying lengths of body at the following speeds:

Length on L.W.L.	Ft. 15	Ft. 20	Ft. 25	Ft. 30	Ft. 35	Ft. 40	Ft. 45	Ft. 50	Ft. 60	Ft. 70	Ft. 80	Ft. 90	Ft. 100	Ft. 120	Ft. 140	Ft. 160	Ft. 200
Speed in knots	4·0	4·5	5·0	5·5	5·9	6·3	6·7	7·1	7·8	8·4	9·0	9·5	10·0	11·0	11·8	12·7	14·0

The speeds in the above table denote the period when a yacht of given length might be expected to meet with an increase in the ratio of the resistance; and it partly explains why, sometimes with only a small accession of wind, a larger sailing yacht will commence to gain much faster than she has hitherto been doing on a smaller yacht. Say the speed of a 60ft. vessel has been 7 knots and that of an 80ft. 7·5 knots, then, if the wind increased so as to drive the former to eight knots, she would arrive at speed when she would experience an augmentation in the ratio of the growth of her resistance; but the 80ft. vessel could put on this addition of one knot to her speed without experiencing any such increase, and the ultimate effect would be that her speed would be accelerated in its relation to the speed of the 60ft. yacht.

For any given displacement, deepening (see pages 131, 145), by adding to the depth and diminishing the breadth, will tend to make the speed higher at which there will be an increase in the ratio of the growth of the resistance; on the other hand, the growth when it does commence will be greater, and the hump in the curve will consequently be more pronounced. The reason why the speed should be higher is accounted for by the fact that the transverse waves become longer or the crests more separated by removing farther forward and aft as the depth of action increases.

Making the ends finer and the mid-section larger, in any given dis-

placement, tends to cause the hump to appear at lower speeds, but in a less pronounced form. The reason of this is that the bow wave is brought aft, and the stern wave forward, so that the transverse wave becomes shorter.

It will thus be seen that the growth of the resistance is affected at some particular speed by the position of the transverse waves on either side of the vessel, and these positions may again vary if the vessel is inclined. It has already been intimated (see page 145) that U sections in the bow and V shape sections in the stern may tend to reduce the general resistance, and the reason seems to be the part the after body plays in making transverse waves is all waste power, whereas some of the power expended may, under certain conditions, lessen the resistance due to the diverging wave-making; some of the effect of U sections in the bow, which deepen the transverse wave, may be recovered by the manner it affects the after body, and Mr. R. E. Froude, in explanation of this matter, says: *

"To destroy or absorb the bow-wave system must require the action of forces derived somehow from the arrangement of stream-line pressures proper to the advance of the after-body through the fluid; which arrangement of pressures is in itself the source of the independent wave-making action of the after-body. Since then, both the absorption of the bow-wave and the foundation of the stern-wave spring from the same source, it is impossible that the introduction of the former operation should not affect the latter. The bow-wave in being absorbed must do something to the stern-wave. It suggests itself as a reasonable hypothesis that this action is to arrest its formation. That, in fact, the function of the after-body when advancing into water already in a certain state of wave-motion, is to swallow up that wave instead of making any of its own. If so, the placing of the after-body in the most favourable position in reference to the bow-wave series has a double benefit; (1) the bow waves restore their energy and are absorbed; and (2) in doing so they prevent the expenditure of energy in making stern waves.

"On this view the theoretically perfect result (*i.e.*, which would be obtainable with the best position of after-body if the two ends of the ship were alike, and if the bow-wave series did not spread away sideways before reaching the after-body), would be that there would be no transverse waves left at all, and no resistance due to their formation.

"Applying these propositions to the wave-making of a ship they amount to this, that the combined or resultant wave series left astern of the ship will be such remainder of the bow-wave series as would have been there but for the after-body (and which may be called the bow component),

* *Vide* "Transactions Institution Naval Architects," 1881.

'super-posed' upon what may be called the 'natural' stern-wave series, *i.e.*, the series which would have been made by the after-body if there had been no remainder of the bow-wave series (and which may be called the stern component).

"And, therefore, when the two sets of crests coincide the resultant wave height will be greatest, and will equal the sum of the heights of the two components; and when the crests of one component coincide with the troughs of the other, the resultant wave height will be smallest, and will equal their difference.

"It has been pointed out that the height of the waves made, and the amount of the resistance caused, will be of the maximum or minimum, according as the crests of the bow-wave series coincide with the crests, or troughs, of the natural stern-wave series. It follows also from the theory that in either of these two cases the crest of the 'resultant wave' coincides with the crest of the larger of the two components, while if the crests of one series fall on the slopes of another, the 'resultant' crest position will be a compromise between the crest positions of the components, though nearer, of course, to the larger of the two.

"The maximum resistance and largest wave will be, as the theory prescribes, where the crest positions of the components coincide, and the minimum resistance and smallest waves where the crest of one falls in the trough of another."

Mr. Froude, by aid of the foregoing observations, demonstrated that the "hump" would occur in the resistance curve, or, in other words, that there would be an increase in the growth of the resistance, when the vessel moved at such a speed that the crests of the bow and stern waves coincided, and a decrease when the crest of one coincided with the trough of the other.

It would appear that the ratio of the growth of the resistance would be at a minimum when the transverse wave crest is at or about the middle of length of the after body. This, of course, does not mean that if of two vessels one (A) has a wave crest at the point mentioned, and the other (B) at a point further aft, that the former will have the least resistance, or, in other words, be the faster boat. Such is by no means the case. The meaning is, that if A were driven at some other speed, so as to draw the wave farther aft, she would then have an increase in the ratio of the growth of resistance, similar to that which B had experienced, and a "hump" would appear in her speed curve; on the other hand, if B were now driven yet faster, she would bring her wave still farther aft and finally get over her hump and some way up the steep incline shown by R S (Fig. 88). The humps and hollows in speed and resistance curves occur, therefore, as the transverse waves shift their positions, relative to the vessel, at varying speeds.

Torpedo boats, when they travel at high speeds and surmount the steep part of the curve, R S, bring the crest of the transverse wave abaft the stern, but it would be impossible for a sailing yacht of similar length (say 100ft.) to do this. As the crest goes aft there is, consequently, a trough under the quarter into which the stern of the boat sinks, hence it has been found an advantage in torpedo boats of 100ft. length at high speeds to move their weights forward. The explanation of this is that by depressing the bow the action in forming the transverse wave is carried forward, and, as a consequence, the wave itself must come forward, bringing a wave crest under the quarter instead of a wave hollow.

It should also be noted that whilst the usual bow divergent wave is thrown off from the bow of torpedo boats, a second wave is formed, on the

FIG. 89.

back slope of which the bow rises at the highest speeds, and, although the stern may sink into the hollow aft, there is relatively to the normal water level a sensible lifting of the boat. (See Fig. 89, which is a copy of a photograph taken from a high speed launch built by Messrs. Simpson and Strickland, of Dartmouth).

RESISTANCE VARIED BY ALTERATIONS IN DISPLACEMENT.

If a vessel of a given length, breadth, and depth has a certain displacement, and if another vessel of similar dimensions has a larger displacement, it may follow that the latter has the greater resistance; but this may only be true at high speeds, when the total resistance is largely made up from wave making; as the fuller form of the larger displacement may yield a smaller skin surface for friction, and thus have an advantage at low speeds.

If the yacht of given dimensions is sunk deeper in the water by the aid of ballast, or other means, the resistance for any given speed will

probably be added to in a ratio with the $\frac{2}{3}$ power of the displacement; but in a sailing yacht there might be an ultimate gain if her stiffness or power to carry sail has been increased by the addition of ballast. By so sinking the vessel in the water the area scale of the curve of sectional areas would be increased, and the augmentation of the resistance would be due to the emphasised transverse wave-making (see pages 145-6).

With regard to differences in resistance at moderate speeds due to alteration in immersion or displacement, it is safe to attribute the variations to the alterations made in the area of immersed surface; and thus any advantages which might accrue from making a small reduction in the weights of a vessel for sailing in light winds depend chiefly upon a sensible decrease being made in the area of immersed surface. Thus, about 4 tons of ballast, or other weight, would require to be removed to lighten the 64ft. cutter Isolde 3in., and 4 tons would be 6·6 per cent. of her displacement. There would be a reduction of about 33 sq. ft. in her area of immersed surface, or about 2·8 per cent. of the whole. At moderate speeds up to 5 knots the gain per hour might be as much as 0·1 mile, or about one mile in fifty, or twelve minutes in ten hours; and in very light winds the gain would be still more marked. No doubt this would be a very important advantage to obtain; but if the wind increased to a pressure of from 0·1lb. to 2lb. per square foot of canvas, which would involve an inclination of from 20° to 30°, the case would be entirely altered, as weight, in the form of ballast low situated, cannot be removed from a vessel without reducing her stability, and thereby reducing the effectiveness of her canvas. A vessel always ought to be provided with the greatest possible stiffness and stability compared with other competitors to meet the wind pressures just indicated. It follows, however, that if a vessel can be lightened by reducing the weight of her gear, spars, or fittings, there will be a gain in speed, due to a probable increase in stability and to the positive decrease in weight and immersed surface.

With regard to an addition to the displacement by increasing the spacing between the water-lines and maintaining their original areas, the Freda and Vanessa in a general way afford an illustration of what can be done in this way. The displacement of Vanessa is 28·5 tons, and that of the Freda 35·5 tons. The 7 tons extra displacement were got into the Freda by a general deepening of her body, as illustrated by Fig. 90.

The character of the distribution of the displacement in a fore and aft direction remained nearly the same, as shown in Fig. 91.

As the mid-section area of Freda is 45 sq. ft., and that of Vanessa 36 sq. ft., the area scale of Freda would be the larger in the ratio of the differences in their relative mid-section areas.

At the first thought it might appear that adding 7 tons to the displacement of a 20-tonner, by general increase in her depth, would so seriously influence the resistance that the yacht would fail in point of speed. However, it would seem that the resistance due to displacement, by adding to the depth of immersion, increases in a less ratio than the theory prescribes at moderate speeds, as increase of depth does not much affect the divergent wave-making at the bow and stern, although it does the transverse wave-making developed at the higher speeds.

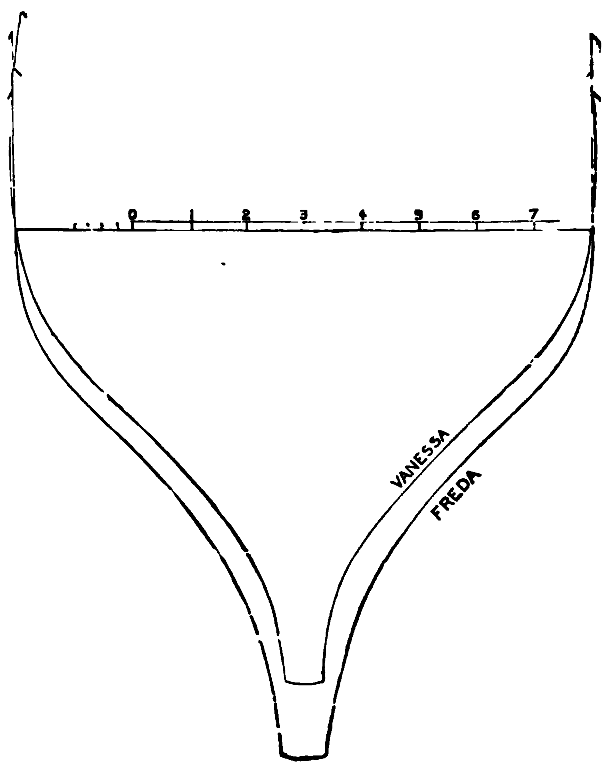


FIG. 90.

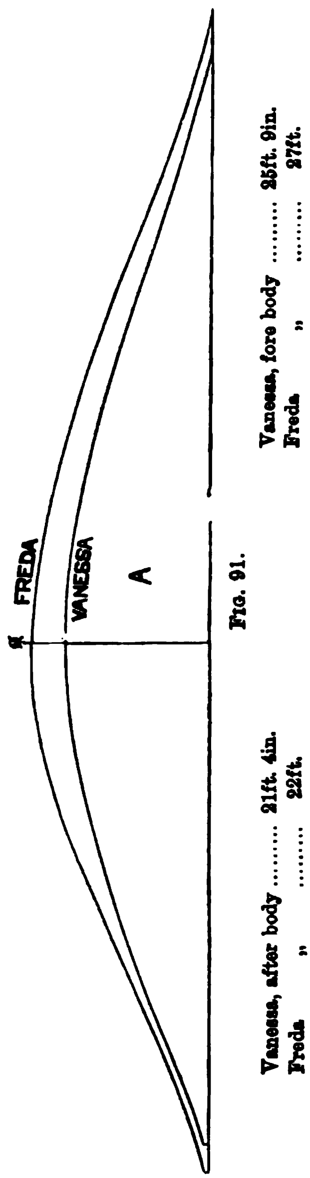


FIG. 91.

Still, the resistance due to the extra 7 tons, as compared with the resistance of Vanessa, could not have been overcome had the yacht been ballasted as Vanessa was. The latter had only 2·5 tons of lead on her keel, of a total weight of ballast of 16·5 tons, whereas Freda had her total weight of 19 tons on her keel; and the total area of sail of Freda was 3140 sq. ft., and 2620 sq. ft. for Vanessa. By the formula given on page 101, and taking the speed

at 8 knots, and the resistance as the cube of the speed, the resistance of the Freda due to the 7 tons displacement would be exactly met by her additional sail spread, so the advantage so far would not be exhibited in speed; but the increase of stiffness due to extra weight and the lower position of her ballast, and the advantage of 2ft. increased length of water-line formed no part of the estimate; but it is obvious that they would give her a considerable advantage in the matter of speed. In the case of a steam yacht the extra resistance due to the increased displacement could, of course, be overcome by extra steam power, which would be a matter of simple calculation.

It must be understood that there was a great step between the ballasting and canvassing of Vanessa and Freda, and, so far as competitive sailing is concerned, Freda obtained the advantages recited without being taxed for the same under the then prevailing tonnage rule. Under the existing rating by length, breadth, girth, and sail area, a yacht of Vanessa's length could not have her length and sail area increased without being taxed, and, as a consequence, there is not, under existing rating rules, an advantage in increasing displacement, so far as competitive sailing is concerned. In short, any rating rule which in any way imposes a tax on sail area must necessarily also impose a tax on displacement, so that the body to be driven will be kept as small as possible.

The advantage to be gained under certain conditions in increasing the displacement by opening the spacing between the water-lines has already been explained; and the matter of increasing the displacement by spacing out the transverse sections can now be considered. Thus, say a yacht of 45·4ft. on L.W.L., 28·9 tons displacement, had to be lengthened out so as to increase her displacement to 35·5 tons, and retain intact the breadth and depth and the areas of the respective cross sections, the length of the yacht would have to be increased 10·4ft., bringing it up to 55·8ft. The nature of the increase on the stability which could be expected under such conditions is explained on page 45, assuming that the vertical position of the centre of gravity remained unchanged by a re-arranged ballast plan. Under such conditions the stability would admit of the sail spread being increased from 2641 sq. ft. to 3130 sq. ft. Of course an increase of speed would be looked for, but this would scarcely be realised in light winds, on account of the large addition which would have been made to the immersed surface, The wave-making resistance at and above 8 knots would be less, and the yacht could be driven on to a higher speed before the wave-making became very formidable (see page 145). It can thus be seen that to attempt to calculate the sail spread from $D^{\frac{2}{3}}$ alone (see page 154) for an increase of displacement by lengthening as described might be very misleading, even

in vessels which have curves of displacement of a similar character if the area scales of the curves vary.

Beyond what has been said as to the advantages or disadvantages of lengthening as just explained, the fact must not be overlooked that the rating of the yacht would be increased by nearly 40 per cent.

In the case of a steam yacht of 45·4ft. L.W.L., with power to drive her 8 knots, no increase of speed could be expected by increasing the spacing between the sectional areas, for the same indicated horse power; in fact, there probably would be a loss of speed on account of the large additions made to the immersed surface and displacement; but if it were contemplated to drive the yacht, say up to 10 knots, by adding to her steam power, it would be better to lengthen her by increasing the spacing and raising the displacement from 28·9 tons to 35·5 tons, than to add either to her depth or breadth; and better than to add jointly to her length, breadth, and depth, so as to make the required addition to the displacement, for the reason that the wave-making resistance with the longer vessel would be increasing at a lower power—perhaps less than the cube—of the speed due to the longer features, which were implicated in making the waves.

The case of lengthening *without* increasing the displacement has already been referred to (page 145).

We have now to consider the question of raising a yacht's displacement from 28·9 to 35·5 tons by increasing all the dimensions and retaining the form of the sections, and consequently the form or character of the curve of areas. The new dimensions E would be obtained from

$$E = D \times \sqrt[3]{\frac{35\cdot5}{28\cdot9}} = \sqrt[3]{D^3 \times \frac{35\cdot5}{28\cdot9}}$$

D being the dimension to be enlarged, and E the enlarged dimension. By this equation the length of a 45·4ft. yacht would be increased to 48·62ft.

In this case the resistance at low speeds, say 4 to 5 knots, would depend upon the addition made to the immersed surface, and the canvas required for any speed would be determined by the formula on page 98.

For unequal speeds—say the speed of each is to be proportioned to the square roots of their dimensions, then the resistances will be as the cubes of their dimensions.* If their full speeds are to be 8·42 knots for 45·4ft. length and 8·71 knots for the enlarged yacht, then, if we put the resistances in terms of the sail spread we have (see also page 101 for sail spread)

$$\text{Sail spread} = \frac{\text{Sail} \times (l^3)}{L^3} = \frac{2641 \times (48\cdot62^3)}{45\cdot4^3} = 3244$$

That is, it would take 3244 sq. ft. sail area to drive the enlarged yacht

* See Mr. Froude's paper in the Transactions of the Institution of Naval Architects, 1874.

at the higher speed due to her length, assuming the wind pressure to be constant. This of course does not take into account any extra stiffness which might be given to the enlarged yacht, beyond that due to her increased displacement and dimensions (see the stability calculations farther on).

It must be pointed out that this method of comparison would be inapplicable for yachts of greatly varying size. For instance, a 45·4ft. lengthened to 84ft., with the beam and draught increased in the same proportion as the length, would have a displacement of 200 tons, and a corresponding sail spread of 16,600 sq. ft.

We have yet to consider the effect of introducing a parallel length of middle body on the growth of the resistance.

Mr. Froude says that alterations of the character of the curve of areas so far as it partakes of the nature of introducing a parallel middle body, the total length being increased by the amount of quasi-parallel body introduced will have the effect of emphasising the hump in the curve and bringing to light other humps at higher speeds (see Fig. 88); and the position of all these humps in the speed scale will depend on the equivalent or effective length or parallel middle body.

It is a part of the wave-line theory already referred to that between a given wave-formed bow and stern any reasonable length of parallel middle body could be introduced without affecting the resistance further than that due to an increase of skin friction and displacement; but it will be shown that this is by no means an exhaustive view of the effect of middle body on resistance, and we will briefly recount the results of Mr. Froude's experiments in that direction, the bow and stern remaining the same.

	Length of Ship. Feet.	Beam. Feet.	Draught. Feet.	Length of Middle Body. Feet.	Displacement. Tons.
A	160	38·4	14·4	00	1245
B	200	„	„	40	1814
C	240	„	„	80	2383
D	260	„	„	100	2667
E	280	„	„	120	2952
F	480	„	„	320	5789

The total resistances of these models, with varying lengths of middle body,* were as shown in the diagram Fig. 92. It will be seen that up to eleven knots, adding to the length of body adds to the resistance in an almost constant ratio; but at speeds above eleven knots the vessel with 40ft. inserted shows a rapid growth of resistance, and just past twelve knots crosses the curve of the ship which has 80ft. of middle body.

Above fourteen knots the vessel which has 120ft. of middle body

* By "length of middle body" is meant that for a certain distance amidships the cross sections are exactly alike, and not merely that the load water-line shows a straightness or flatness amidships.

begins to come down in growth of resistance, and at fifteen and a half knots her total resistance would probably only equal that of the 160ft. ship, which has no parallel length of middle body at all.

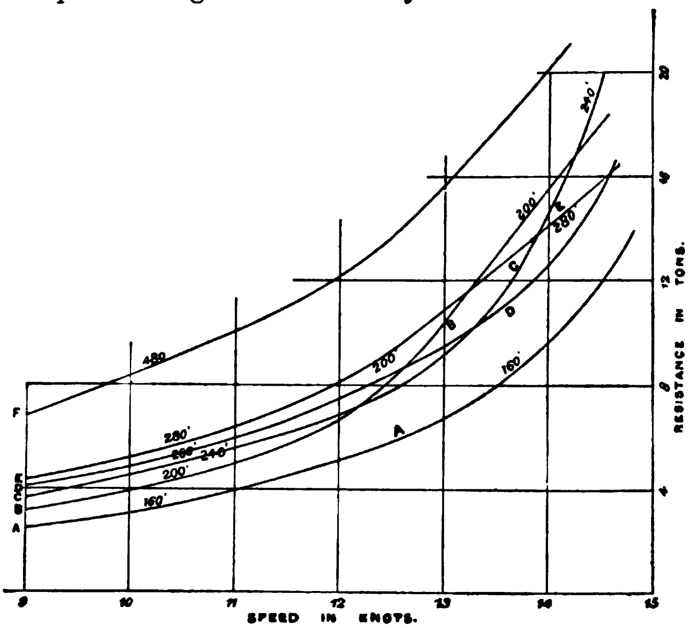


FIG. 92.

It was certainly a most extraordinary discovery to make, that at some higher speed a ship 2383 tons displacement should have actually a smaller

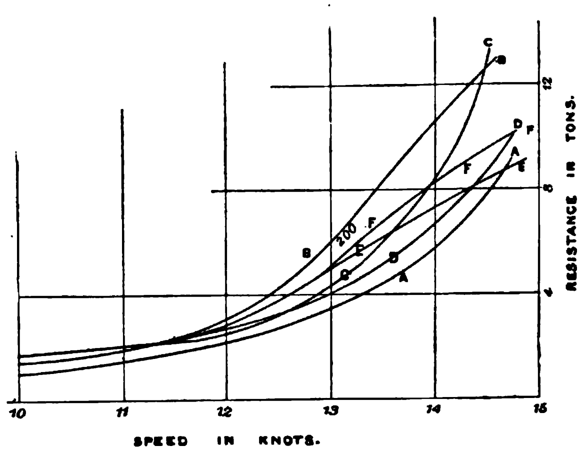


FIG. 93.

resistance than one of 1814 tons displacement, although the ends of each were exactly alike. It is curious to note, too, how the resistance curves of these two ships (see 200ft. and 240ft. in the diagram) cross each other again

after reaching fourteen knots, their resistances being actually equal at 12·3 knots and 14·2 knots. A still more striking feature is the fact that the ship of 280ft. length and 2952 tons displacement has only the same resistance as the 200ft. ship of 1814 tons displacement at 13·3 knots, and at 14·5 knots has lowest resistance of all, excepting the 160ft. ship, which has no parallel length of middle body.

To account for these peculiarities, Mr. Froude eliminated from each model the resistance due to surface friction, and produced curves of the resistance which remained, which he termed "residuary resistance." This resistance was of course due to wave making, and the curves formed for the resistance at different speeds are shown in Fig. 93. It will be seen how small is the resistance of the 280ft. ship, with 120ft. length of straight middle body, compared with the 200ft. ship with only 40ft. of middle body. This advantage was not, however, maintained if another 40ft. were added; but reappeared (at a higher speed) if yet another 40ft. were added, making the ship 360ft. long.

From a series of observations made on the models when running at varying speeds, Mr. Froude was able to measure and trace the positions of the transverse waves relative to the side of the ship for each speed and each length of middle body. He found that a series of crests occurred at successive distances from the bow, with troughs between them, as explained on page 130. A general conclusion was, that a speed which caused a wave trough about the mid length of the after body had an accession of resistance, and a wave crest at the same point brought about a decrease in the growth of the resistance, as there was then an increase of what Mr. Froude termed "quasi hydrostatic pressure" against the after body.

This, however, was only a general conclusion, and the effect of the combined action of transverse waves and stern waves in varying the growth of resistance will be found referred to on page 152.

The foregoing experiments will be more useful by way of comparison, if the vessels are reduced to sizes appropriate for yachts, which can be done by taking one-fourth the dimensions; the lengths of the yachts and their displacements will then be as follows:

	Total Length of Yacht.	Length of Straight Middle Body.	Displacement.
	FT.	FT.	TONS.
A	40	00	19·4
B	50	10	28·3
C	60	20	37·2
D	65	25	41·7
E	70	30	46·0
F	120	80	90·5

The corresponding speeds for the resistances shown on Figs. 92 and 93 will be

For Ship............Knots	9	10	11	12	13	14	15
For Yacht ....Knots	4·5	5·0	5·5	6·0	6·5	7·0	7·5

Or take yachts half the dimensions of the ships:

	Total Length of Yacht.	Length of Straight Middle Body.	Displacement.
	FT.	FT.	TONS.
A	80	00	155
B	100	20	227
C	120	40	298
D	130	50	333
E	140	60	369
F	240	160	723

The corresponding speeds for the resistances (Figs. 92 and 93) will be

For Ship............Knots	9	10	11	12	13	14	15
For Yacht ....Knots	6·5	7·1	7·8	8·6	9·3	10	10·7

From the foregoing it can be inferred that if a desire existed to lengthen an 80ft. steam yacht, it would be more economical, so far as engine power goes, to put 40ft. or 60ft. into the middle of her rather than 20ft. if the yacht had to be driven at any speed a little above 8·6 knots (see B curve, Figs. 92 and 93); but the resistance begins to fall again after 10 knots are reached for the yacht with 20ft. inserted. In connection with this matter it is worthy of note that the Xantha steam yacht, of 105ft. length on load water line, was lengthened amidships 20ft., and the speed only fell from 11·8 to 11·2 knots, with exactly the same engine power.

These comparisons will be pretty exact so long as the dimensions of bow and stern and middle-body are proportioned to those given, no matter what the form, and in a general way the character of the curves will answer for proportion and displacements.

CHAPTER IX.

RESISTANCE EXPERIMENTS WITH MODELS.

A GREAT many attempts had been from time to time made with models with a view of determining the form of least resistance, but as the models were seldom of "ship-shape form," and as their speeds were made to compare directly with the speeds of the ship, the results obtained were quite useless. To the late Mr. William Froude the credit is due of discovering the law for comparing the speed of a ship with that of her model. The discovery made by Mr. Scott Russell that a vessel could be driven at a speed equal to the square root of the total length of bow and stern without undue wave-making resistance indicated that, to render model experiments of value, the speed of the model should be suitable to her length; she would then have similar wave-making characteristics or configuration to the ship, and the resistances would be comparable. Mr. Froude proved that this was actually the true state of the case except in the matter of friction. That is, to compare the model of the ship with that of the ship itself, the model must only make the speed due to her own length; that is to say, if the speed of the ship was as the square root of her length, that of the model must be only as the square root of the model's length; and for other speeds, greater or smaller than the square root of the length, the corresponding speed would be as the square root of the scale of comparison. By scale of comparison is meant the proportion the model in dimensions bears to the dimensions of the ship. If the length of the ship (L) be 81ft. on the water-line, and the model (l) 3ft., then the model will be $\frac{1}{27}$ the dimensions of the ship ($\frac{L}{l} = \frac{81}{3} = 27$), and this will be the scale of comparison. Next, if the model is timed to make a speed at the rate of say 2·25 knots an hour, then the speed of the ship (S) will be found by multiplying the speed of the model, s (2·25 knots), by the square root of the scale of comparison.

$$S = \sqrt{\frac{L}{l}} \times s = \sqrt{\frac{81}{3}} \times 2{\cdot}25 = 5{\cdot}2 \times 2{\cdot}25 = 11{\cdot}7 \text{ knots.}$$

(The square root of 27 is 5·2, and 2·25 multiplied by 5·2 = 11·7 knots.)

Thus, if it were sought to ascertain which of several models compared the most advantageously in point of resistance at some given speed for the ship, say 11·7 knots, they must be towed at the "corresponding speed," which it will be seen would be 2·25 knots in this case. (The speed for the model would be found by *dividing* the speed of the ship by the square root of the scale of comparison. $\frac{11 \cdot 7}{5 \cdot 2} = 2 \cdot 25$ knots).

The actual resistance of a ship in pounds can also be computed from the resistance of her model, although the calculation is likely to involve considerable error in consequence of the variation in the intensity of the surface friction for different lengths in spite of a correction for surface friction. The smaller the model the greater the error is likely to be.

The resistance of the ship to that of the model will be as the cube of the scale of comparison. Thus, assuming the model to be on a scale $\frac{1}{9}$ that of the ship, the cube of 9 is 729; therefore the resistance of the ship for *any speed* will be 729 times the resistance of the model at the "corresponding speed."*

For example, assume that a model $\frac{1}{9}$ the dimensions of the Sappho be made, the length of such model would be 13·5ft. Next assume that at a speed of 3 knots it was ascertained by experiment that the resistance of such a model was 6lb.; then the speed of the Sappho for the "corresponding speed" of 9 knots would be 6lb. × 729, or 4374lb.

But, inasmuch as the mean value of the surface friction *decreases* in proportion as the length is increased, a correction has to be made to arrive at the *actual* resistance of the ship (see page 119).

The resistance for varnish on a surface which has a length of 13ft. in the direction of motion is 0·3lb. per square foot, and the resistance varies as the 1·84 power of the speed.

The resistance for copper on lengths of 100ft. and upwards is 0·25lb. per square foot of surface.

The immersed surface of the Sappho is 3790 square feet; so her surface resistance due to skin friction at 6 knots will be 3790 square feet × 0·25lb. = 947lb. The resistance due to surface friction at 9 knots will have increased as the 1·84 power of the speed.†

The 1·84 power of 6 is 27, and the 1·84 power of 9 is 57; then $\frac{947 \times 57}{27} = 1999$lb., the resistance due to surface friction at 9 knots.

* See Mr. Froude's report to the Admiralty on the Greyhound experiments, published in *Engineering*, May 1, 1874.

† The 1·84 power, or any power, of numbers are thus obtained: Say the speed is 6 knots then the logarithm of 6 is 0·778151, which, multiplied by 1·84, equals 1·4817, which is the logarithm of 27, and 27 is the 1·84 power of 6. In like manner the 1·84 power of other numbers or speeds will be computed.

The surface friction of the model at the "corresponding speeds" would be ascertained by accurately measuring its immersed surface. In this case, as we know the surface of Sappho, that of her model can be readily computed thus: $\frac{\text{Immersed Surface.}}{\text{Scale}^2}$ That is, the immersed surface is divided by the *square* of the scale of comparison. The Sappho is nine times the size of the model, and $9 \times 9 = 81$, therefore the immersed surface of the model in square feet will be $\frac{3790}{81} = 46{\cdot}8$ sq. ft.

The resistance for clean varnish on a surface 13ft. in length is equal to 0·3lb. per square foot at a speed of 6 knots; therefore the resistance for surface friction of the Sappho's model would be $46{\cdot}8 \times 0{\cdot}3 = 14$lb., if moved at 6 knots; but the "corresponding speed" of the model to that of the Sappho is 3 knots, and the frictional resistance due to that speed will show a decrease as the 1·84 power of the speed. The 1·84 power of 6 is 27, and the 1·84 power of 3 is 7·5; then $\frac{14\text{lb.} \times 7{\cdot}5}{27} = 3{\cdot}88$lb., or the resistance of the model due to surface friction at a speed of 3 knots is 3·88lb.

The resistance for the ship for surface friction at the corresponding speed of 9 knots should be $3{\cdot}88\text{lb.} \times \text{scale}^3 = 3{\cdot}88 \times 9^3 = 3{\cdot}88 \times 729 = 2828$lb. But we have already calculated that the resistance of the ship due to this cause at 9 knots is 1999lb.; so the error for surface friction to be read off from the total resistance (4374lb.) calculated for the ship from the model is the difference $2828 - 1999 = 829$lb. Then $4374\text{lb.} - 829\text{lb.} = 3445$lb.; or the total resistance of the Sappho deduced from her model would be 3445lb. at a speed of 9 knots; and 58 per cent. of this resistance would be due to surface friction alone.* It will be gathered from the foregoing that there is a large opening for error in attempting to deduce the actual resistance of a ship from that of her model; but for making comparisons between models such experiments are of great value if the "corresponding speed" be observed.

The resistance at various speeds can be graphically shown by a curve. Ordinates must be ruled at convenient distances at right angles to each other, as shown in Fig. 94. Let the upright ordinates represent the speed, and the horizontal lines the resistance in pounds. Say the total resistance for the model is 6lb. at a speed of 3 knots; this resistance represented in length will be set off on the ordinate erected for that speed. Points on the other ordinates will be obtained as already described, and when set off the curve of resistance A A for the model can be swept in. (The curve shown is an imaginary one, the points for the ordinate at the

* It must be understood that the resistance of a model of the Sappho, at a speed of 3 knots is an assumption introduced to show how the calculations are made.

3 knot speed were, however, calculated from the assumption that the resistance of the model at 3 knots would be 6lb.) The curve B B, shown by the dotted line, represents the resistance of the model due to surface friction alone. Then the surface friction for the ship is obtained as already described, and set off *from* B B *towards* D D, and *not* from the base C C; the vertical distance between the curve A A and the curve D D will then represent the actual resistance for the ship. If the experiment be made in fresh water the resistance will have to be increased for salt water in the proportion of the density of salt water to fresh.

If the experiment is being made to test the speed of a trading vessel it is incumbent that she should be towed at various angles of heel. Any

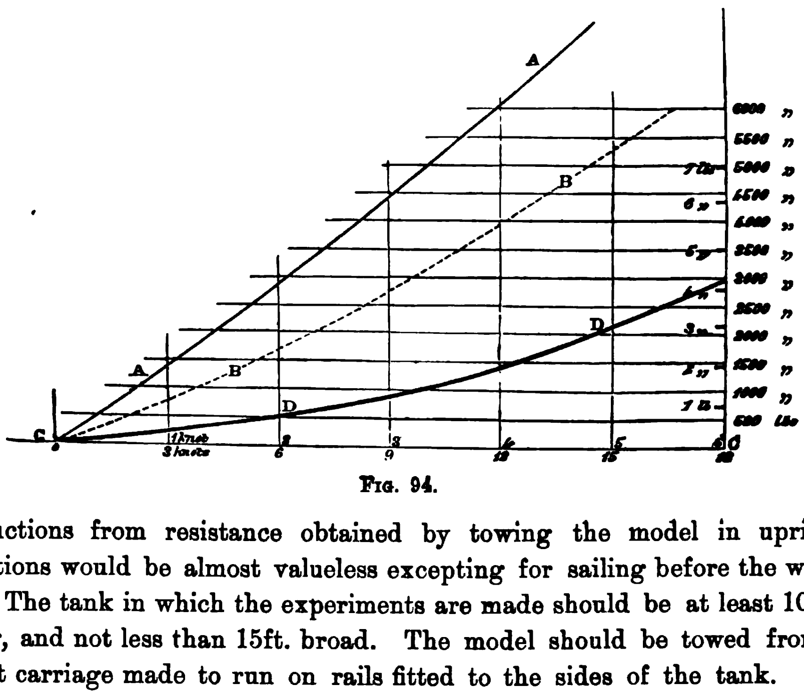


Fig. 94.

deductions from resistance obtained by towing the model in upright positions would be almost valueless excepting for sailing before the wind.

The tank in which the experiments are made should be at least 100ft. long, and not less than 15ft. broad. The model should be towed from a light carriage made to run on rails fitted to the sides of the tank. The towing line would be attached to a dynamometer carried on the carriage. *The speed throughout each run must be uniform.*

If it is desired to get the *actual resistance* of the ship in pounds from that of the model, the model should be at least 8ft. long; but the relative speeds of yachts could be ascertained from their models of smaller size, providing they are towed at the "corresponding speeds" as described.

CHAPTER X.

THEORY AND PRACTICE; THE FORE BODY; THE AFTER BODY.

In this chapter we shall endeavour to determine the extent to which theory as to form and general practice as to form agree. Wave lines closely approximating to curves of versed sines have usually appeared in the fore body of English and American yachts; but, so far as racing yachts are concerned, they gradually disappeared in this country under the influence of the old tonnage rule, which, as already explained, forced designers into constructing extremely narrow sailing yachts, the beam in many cases not being more than ·17 of the length, or, say, one-sixth the length. In such narrow yachts the angle of the entrance would naturally be very fine, and for that reason alone required no assistance from wave line features to ease the entrance. Beyond this, hollow lines in the bow of very narrow vessels would not admit of sufficient buoyancy in the fore end for good performance in rough water, and for that reason would be objectionable. The practice of much cutting away the fore foot also assisted in the disappearance of hollow lines in the entrance, as already explained on page 147. The practice just referred to still prevails, and even now that racing yachts are being built with a beam ranging from ·22 to ·32 of the length, hollow lines in the entrance are not in much favour; if, however, a yacht is given a very deep fore body some hollow in the under water-lines would be in most cases indispensable.

The American yacht Sappho (as given among the list of Plates) exhibits in a very striking manner a yacht built in accordance with the wave line theory. The relative lengths of her fore and after body are precisely those required for the wave line theory; that is to say, the fore body is one-third longer than the after body. The curves of the water lines of the fore body approximate very closely to curves of versed sines; and the curves of the water lines of the after body are cycloidal or trochoidal. The curve of the load water line in the fore body is not strictly

a curve of versed sines, as it is a little fuller. But the second water line struck two feet below the load water line is an *exact* curve of versed sines; and in a like manner the curved outline of any horizontal section cut off any distance below the second water line is a curve of versed sines. Thus, it can be taken that the Sappho's fore body was modelled according to the requirements of the wave line theory. In the after body the load water line is cycloidal, but the horizontal sections struck below the load line are trochoidal; and the curved outlines of the vertical longitudinal sections (shown by the buttock lines) are parabolas.

In a case like the Sappho's the displacement would be necessarily disposed of longitudinally in accord with the wave form theory.

There has not been such a thorough example of the wave line theory among British yachts, but the Lulworth (designed and built in 1857 by Mr. Joseph Weld, of Lulworth Castle) bears a close resemblance to that theory in her water lines, and she was probably constructed with the intention of testing its value.

So far as the "wave form" of displacement goes, it appears that it has been approximately conformed to since the introduction of Mr. Scott Russell's wave-line theory, as to carry out the one, even in a modified degree, almost necessarily involved the presence of the other. On Plate II. there are depicted the curves of sectional areas of the schooners, Sappho, 232 tons displacement; Livonia, 215 tons; Cambria, 167 tons; Egeria, 142 tons; and cutters, Bloodhound, 47 tons; Lily, 13 tons; Lulworth, 63 tons. The displacement curves of several other more modern yachts will be found on Plates III. and IV. In these exemplary curves the dotted lines show the true wave curve; and in all cases 4 per cent. is added to the length of the fore body and 4 per cent. to the length of the after body before dividing the line into the necessary interval for plotting the curve. This is done to dispose of the extreme fine end of the wave curve.

It will be seen that there is a general agreement with the wave curve, and it should be noted that the majority of the vessels, at least, were designed without any reference to the wave-form theory. Most of the entrances are a trifle fuller than the wave form prescribes, and some have the fulness of a trochoid; but this fulness is not apparent in modern racing yachts. The fin-like lead keel, bulky in its mid length and tapering fore and aft, has the effect of adding to the areas of the midship cross sections; and again the areas of the sections in the fore part and in the stern part are diminished by the upward rake of the keel towards the stem and the great rake given to the stern post aft; the general effect on the curve of

sections is that it is finer than the wave-form theory prescribes, and the yacht concerned appears to have a very large mid-section and fine ends, characteristics which were regarded as highly objectionable a few years ago. But the fact is, that the ends of the curve of displacement are only finer because a useless weight of solid wood has been trimmed away both forward and aft; and the bulky lead keel fin, whilst destroying the beautiful symmetry of the wave curve, compensates for its defects in this respect by the great influence it has on stability. In the case of a vessel like the 30ft. linear rating yacht Dolphin, fitted with a deep fin-like keel amidships, the effect is, although the keel is of no great thickness, to produce a kind of hump on the back of the curve of displacement; but this is only a more pronounced effect of the influences already referred to as having destroyed the symmetry of the wave curve. If the fin-like keel were removed, and the areas of the sections plotted without it, the curve would become an almost true wave curve.

The raking keel and cutaway fore foot are features which existed centuries ago, and are to be met with in old drawings of ships. George Steers introduced them in 1851 in the famous America yacht; it is also very apparent in the Lulworth; and it was carried to even a greater extent in the Kitten, built by Harvey in the year 1852; but the deep fin, or bulb fin keel, was not in vogue until 1888.

In the case of steam yachts there is a very considerable departure from the wave-line and wave-form theory, although some have been made to accord with it very closely. Notable examples of the latter are the Marchesa, Xantha, Amazon, Fair Geraldine, Linotte, Speedy, Fanvette, and Oriental. It should, however, be borne in mind that many steam yachts have in a varying degree the length of parallel side referred to on pages 159 and 162; it would therefore in some cases be a more trustworthy test to only take the bow end and stern end for exemplifying the displacement curve. Still, there cannot be a question that those steam yachts are most favourably formed for speed which have practically no straight length of middle body, and whose curves of sectional areas closely accord with the wave form. There may, of course, be several urgent reasons for not adopting such a form—such, for instance, as in a vessel of limited draught, where the necessary displacement could only be obtained by the fulness being carried fore and aft into the ends to obtain the required displacement; or for a shallow vessel, with a very heavy top, intended to work in a short steep sea. The placing of the machinery either very far forward or very far aft might also of necessity influence the form given to the displacement. Again, it might be necessary to give the vessel great stowage room for coal for ocean

voyages, and this could not very well be provided except in the middle part of the length. Also, if the vessel is very long in proportion to her breadth and depth, it would probably be necessary that she should have some straight length of middle body to obtain the required displacement. When, however, there is no object in restricting depth, there would be an advantage, both for speed and seaworthiness, by having a somewhat deeper middle body with less length of parallel side.

The steam yacht Linotte illustrates a moderate departure from the wave-form in general fulness in the fore body. In her case a given displacement had to be provided within certain dimensions and limitation of draught of water; and, so far as the fore body is concerned, it would appear, judging from the example of Linotte, that the fulness in the fore body is no great detriment provided the sections are not deep and if the after body is well formed, as she proved to be exceptionally fast with a moderate power.

It will be well now to examine the processes which have led the designers of yachts into such a generally uniform result with regard to the fore and aft disposition of the displacement. The simplest plan to achieve this will be to follow as closely as possible the process of construction, commencing with the body.

The Fore Body.

In considering the fore body, it will be necessary to first determine its length. The position of the midship section practically determines the relative lengths of the fore and after body; but there appears to be no common practice in appointing that position. The case of the Sappho is the nearest approach to the wave form theory proportion, and there are several approximate examples before us, such as the Arrow, Audrey, Isolde, and Kismet, come next in the examples given. We cannot, however, shut our eyes to the fact that equally successful craft in their way, such as Penitent, Florinda, and Vanduara, are very much under that proportion. It would, therefore, appear to be some foundation for the statement, a general result from experience, that it is more important that the after body should be of the length stipulated for any required speed, than that the fore body should. It does not follow, however, that no improvement in speed would accrue if the fore body were also of the stipulated length.

In the following table the proportionate length of fore body to length of load line in several well-known vessels is set forth. $\frac{\text{Fore body}}{\text{L.W.L.}}$ = the proportional length of the fore body to the length of load line. The length

of after body will be the complement of the fractions representing the length of the fore body.

Name of Yacht.	Ratio of length of fore body to length of load line.	Name of Yacht.	Ratio of length of fore body to length of load line.
Wave length theory	·600	Ghost (20-rater, cutter)	·574
Kriemhilda (cutter)	·523	Volunteer (American cutter)	·581
Miranda (schooner)	·533	Sleuthhound (40-tonner)	·583
Dolphin (2·5-rater)	·533	Mayflower (American cutter)	·585
Florinda (yawl)	·535	Sappho (American schooner)	·590
Lily (10-tons cutter)	·538	Isolde (65ft. rating), 1895	·566
Vanduara (cutter)	·548	Penitent (52ft. rating), 1896	·540
Genesta (cutter)	·540	Stephanie (52ft. rating), 1894	·558
Vanessa (20-tons, cutter)	·550	Audrey (52ft. rating), 1896	·577
Constance (yawl)	·554	Kismet (18ft. rating), 1896	·589
Neptune (10-tons)	·557	Marchesa (S.Y.)*	·486
Aline (schooner)	·560	Capercailzie (S.Y.)	·500
Livonia (schooner)	·560	Oriental (S.Y.)	·504
Latona (yawl)	·562	Linotte (S.Y.)	·509
Minerva (21-rater, cutter)	·562	Queen Marfisa (S.Y.)	·492
Egeria (schooner)	·564	Amazon (S.Y.)	·512
Samoena (cutter)	·569	Chazalie (S.Y.)	·526
Arrow (cutter)	·570	Fair Geraldine (S.Y.)	·529

* In the case of steam yachts the extreme length to outer stern post is reckoned.

From this table it would appear that the proportion of length of fore body to total length of load line should be about ·55; or, in other words, that the position of the mid-section should be about ·05 of the length of load line abaft the centre of length. If the length of load line is 80ft., then 80 × ·05 = 4ft., the distance the mid-section is to be abaft the centre of length.

In yachts the general practice is to place the greatest breadth of the load water-line abaft the position of what is known as the mid-section; a larger and finer entrance at the surface is thereby obtained, and the resistance from the diverging bow wave is somewhat modified. This practice has a striking example in the case of the Isolde, Neptune, and Minerva, designed by Mr. Fife, and will be again referred to.

With regard to the form of the water lines, it has already been pointed out that they are usually much fuller than the wave line theory prescribes, especially near the load water plane; and it must always be borne in mind that the greater the area of load water plane in comparison to the area of the lower water lines, the higher the meta-centre will be, and the greater the stability for any given weight and position of weights.

The dividing ordinate at No. 4 station (Fig. 95) can be taken as a rough test of the coefficient of fineness for the fore part of the load

water-line; in a curve of versed sines it is ·5 (see page 122): but in no yacht which we have had the opportunity of examining has the coefficient been as low as ·5, but ranges from ·583 to ·74. In different vessels the ratio of the length of the dividing ordinate, *a a* (Fig. 95), to the length of the ordinate at the greatest half breadth wherever found, A C, on the load water-line is as follows in the table $\left(\frac{a\,a}{\mathrm{A\ C}}\right)$

Name of Yacht.	Value of *a a*.	Name of Yacht.	Value of *a a*.
Sappho (American schooner)	·583	Aline (schooner)	·677
Volunteer (American cutter)	·623	Lenore (yawl)	·682
Minerva (21-rater)	·650	Neptune (10-tons, cutter)	·696
Rose of Devon (yawl)	·656	Boadicea (schooner)	·700
Milly (cutter)	·657	Elmina (schooner)	·700
Constance (yawl)	·660	Hildegarde (schooner)	·700
Bloodhound (cutter)	·660	Kriemhilda (cutter)	·700
Florinda (yawl)	·660	Sleuthhound (cutter)	·700
Flying Cloud (schooner)	·660	Livonia (schooner)	·705
Miranda (schooner)	·661	Isolde (65ft. rating), 1895	·736
Freda (20-tons, cutter)	·661	Stephanie (52ft. rating), 1894	·720
Fiona (cutter)	·662	Audrey (52ft. rating), 1896	·730
Arrow (cutter)	·663	Kismet (18ft. rating), 1896	·730
Butterfly (cutter)	·663	Wenonah (Herreshoff), 1891	·687
Vanessa (cutter)	·663	Xantha (s.y.)	·650
Britannia (cutter)	·666	Amazon (s.y.)	·657
Egeria (schooner)	·666	Chazalie (s.y.)	·662
Dolphin (30ft. rater)	·666	Capercailzie (s.y.)*	·700
Mosquito (cutter)	·666	Chazalie (s.y.)	·700
Latona (yawl)	·670	Queen of Palmyra (s.y.)	·740
Saraband (10-tons, cutter)	·670	Marchesa (s.y.)	·750
Sea Belle (schooner)	·671	Oriental (s.y.)	·757
Ghost (20-rater)	·673	Capercailzie (s.y.)	·765
Mascotte (3-tons, cutter)	·676	Fair Geraldine (s.y.)	·856

* This is taking A C 12ft. farther ahead, thus allowing for 24ft. straight of breadth.

In Fig. 95 let A B be the length of the fore part of the load water-line, from the greatest half breadth on the L.W.L. to the extreme fore end of the stem; A C will be the greatest half breadth at right angles to A B. The

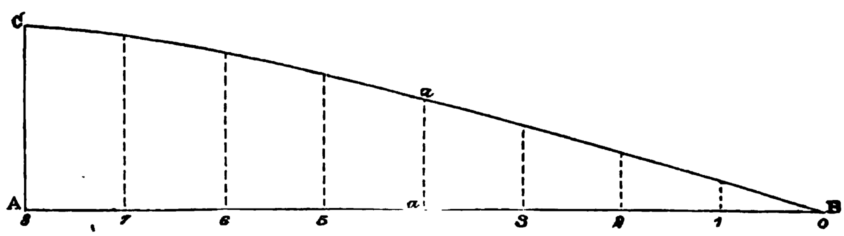


Fig. 95.

dividing ordinate will be at *a a*, midway between A B. From the foregoing data it would seem that ·70 according to the latest type of yacht is a good ratio for this dividing ordinate. Any one practised in yacht

designing could easily sweep in a suitable curve for the fore part of the load water-line, with the two ordinates given, but it will be useful to more minutely define the usual character of the curve. The following factors for such a curve were computed by the author in 1875 as fairly exemplifying the practice at that date, and it has been generally followed since. Divide the length A B (Fig. 95) into eight *equal* intervals as numbered, 0, 1, 2, 3, 4, 5, 6, 7, 8, then the factors for computing the lengths of the ordinates will be:

0	1	2	3	4	5	6	7	8
·0000	·1500	·3200	·5000	·6700	·8125	·9183	·9833	1·0000

That is, suppose the length of the ordinate at 8 (the greatest half breadth) to be 12ft.; then the length of the ordinate at 7 will be 12 × ·9833 = 11·80, and so on, as set forth in the annexed table.

	Greatest half breadth.		Multipliers.		Length of Ordinates.
8.	12ft.	×	1·0000	=	12·00ft.
7.	12ft.	×	·9833	=	11·80ft.
6.	12ft.	×	·9183	=	11·01ft.
5.	12ft.	×	·8125	=	9·75ft.
4.	12ft.	×	·6700	=	8·04ft.
3.	12ft.	×	·5000	=	6·00ft.
2.	12ft.	×	·3200	=	3·84ft.
1.	12ft.	×	·1500	=	1·80ft.
0.	12ft.	×	·0000	=	0·00ft.

The coefficient of such an area will be ·608; thus the area will be the greatest half breadth, A C, multiplied by A B, multiplied by coefficient; or,

$\overline{AC} \times \overline{AB} \times \cdot 608 = 12 \times 40 \times \cdot 608 = 291$ sq. ft. (40ft. being the assumed length A B).

The particular form of curve which these factors would generate is not put forward as absolutely the best that could be devised, but only as fairly representing common practice in vessels of from four to five beams in length.

In the table which follows over leaf, the factors for computing the ordinates of the fore part of the load line in a few well-known vessels are given, assuming the line to be divided into eight equal intervals. The factors were found by dividing the ordinates 1, 2, 3, &c., by the greatest ordinate A B.

This table shows how practice varies with variations of beam, No. 4 column representing the dividing ordinate at the mid-length, as given in the table on page 172. This table on page 172 will be in ordinary practice a sufficient guide when the variation in proportions of beam to length is much about the same as that found in the yachts in the table on page 174.

FACTORS FOR COMPUTING ORDINATES FOR THE FORE PART OF THE L.W.L.

	Breadth/Length.	0 Stem.	1	2	3	4 *a a*	5	6	7	8
Exemplar curve, p. 168	·000	·0000	·1500	·3200	·5000	·6700	·8125	·9183	·9833	1·0000
Aline (schooner).........	·207	·0000	·1531	·3253	·5063	·6773	·8113	·9178	·9810	1·0000
Vol-au-Vent (cutter)...	·217	·0000	·1350	·3255	·5230	·7070	·8553	·9476	·9833	1·0000
Kriemhilda (cutter) ...	·220	·0000	·1620	·3333	·5170	·7000	·8382	·9333	·9842	1·0000
Florinda (yawl)	·223	·0000	·1562	·3230	·4960	·6600	·8020	·9104	·9748	1·0000
Neptune(10-tons,cutter)	·190	·0000	·1800	·3575	·5303	·6960	·8242	·9180	·9850	1·0000
Jullanar (yawl)	·168	·0000	·2160	·4160	·5950	·7260	·8330	·9280	·9760	1·0000
Miranda (schooner) ...	·219	·0000	·1777	·3330	·5022	·6610	·8000	·9080	·9800	1·0000
Lenore (yawl)............	·192	·0000	·1800	·3640	·5400	·7000	·8300	·9300	·9800	1·0000
Genesta (cutter).........	·185	·0000	·1530	·3200	·4933	·6600	·8000	·9000	·9600	1·0000
Ghost (20 rater).........	·217	·0000	·1770	·3540	·5232	·6730	·8100	·9100	·9800	1·0000
Dolphin (2·5 rater) ...	·290	·0000	·1600	·3343	·5000	·6660	·8000	·8943	·9643	1·0000
Minerva (21 rating) ...	·260	·0000	·1600	·3300	·5030	·6504	·7952	·9000	·9650	1·0000
Volunteer (American)..	·267	·0000	·1471	·2941	·4706	·6235	·7706	·8824	·9647	1·0000
Audrey (cutter), 1896	·292	·0000	·2222	·4206	·5825	·7222	·8254	·9127	·9682	1·0000

It will be observed that there is a considerable divergence in the forms of the curves. The coefficient of fineness, or, in other words, the ratio of the area to the circumscribing parallelogram is approximately the same all through in the fore part of the L.W.L., and equals ·608 in nearly all the cases cited; but the case of Audrey illustrates the fulness which can be

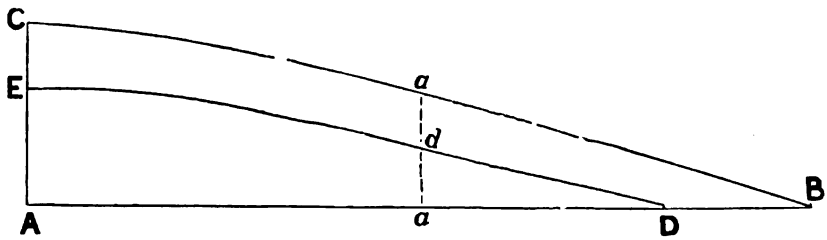


FIG. 96.

successfully given to a vessel of comparatively shallow form. Numerous examples of this will be found amongst the plates. It will be observed how closely the curve of Audrey's load line forward agrees with the Jullanar's curve, the Jullanar representing the extremest of narrow yachts. The curves, which are finest at the fore end, are generally the fullest at the end next the midship section. The Aline's and Miranda's are good examples of curves with a gentle contrary flexure, such as seems suitable to a vessel of 4 or 4½ beams to length, and closely resemble the factors given in the Exemplar curve. The Vol-au-Vent is a fair example of a wave line. It will appear that the Florinda's curve is almost identical with Genesta's, and both closely correspond with the exemplar curve as computed by the author in 1875.

The form of the water lines below the load water-line has next to

be considered. In most vessels the curves of these lines are found about equal in fulness to the curve of the load water line, and this fulness appears to be no detriment when sailing in smooth water at moderate velocities, and, on the whole, an advantage when sailing in rough water. The value of *a d* (Fig. 96) in different vessels for water lines *below* the load water line has been thus computed: the depth, at the balance ordinate *a a* (Fig. 95), was measured from and at right angles to the load water line, to the underside of keel; at *half* this depth a water-line was struck and projected on the half-breadth plan. Then for A B (Fig. 95) we have A D (Fig. 96) the length of this particular water-line from its greatest half breadth to its extreme fore end; but it must be observed that the dividing ordinate *a d* does not occur at the mid length of this line, but at *a a*, the mid length of A B. For A C (Fig. 95) we have A E (Fig. 96) for the half breadth of the midship section at the mid depth below the load water-line; then the value of *a d*, as set out in the table, for this dividing ordinate will be $\frac{a\ d}{\text{A D}}$.

Name of Vessel.	Value of *a d.*	Name of Vessel.	Value of *a d.*
Sappho (American schooner)	·45	Vanessa (cutter)	·54
Egeria (schooner)	·45	Miranda (schooner)	·55
Arrow (cutter)	·45	Butterfly (cutter)	·55
Volunteer (American cutter)	·46	Livonia (schooner)	·55
Vol-au-Vent (cutter)	·48	Sleuthhound (cutter)	·55
Beluga (cutter)	·50	Elmina (schooner)	·56
Genesta (cutter)	·50	Freda (20, cutter)	·57
Kriemhilda (cutter)	·50	Mascotte (cutter)	·57
Iona (cutter)	·50	Lenore (yawl)	·59
Playmate (cutter)	·50	Constance (yawl)	·59
Florinda (yawl)	·51	Hildegarde (schooner)	·60
Keepsake (cutter)	·51	Dolphin (2·5-rater)	·60
Neptune (10-tons, cutter)	·51	Minerva (21-rater)	·61
Saraband (10-tons, cutter)	·51	Sea Belle (schooner)	·62
Cambria (schooner)	·52	Ghost (20-rater)	·63
Aline (schooner)	·53	Audrey (cutter), 1896	·69
Fiona (cutter)	·53	Chazalie (S.Y.)	·51
Latona (yawl)	·53	Amazon (S.Y.)	·54
Milly (cutter)	·53	Capercailzie (S.Y.)*	·59
Mosquito (cutter)	·53	Capercailzie (S.Y.)	·63
Bloodhound (cutter)	·54	Marchesa (S.Y.)	·66
Gwendolin (schooner)	·54	Oriental (S.Y.)	·72
Lily (10-tons, cutter)	·54	Fair Geraldine (S.Y.)	·70

* This is with A C (Fig. 95) put 12ft. further ahead to allow for 24ft. straight of breadth.

It will be seen that there is a considerable range in these values of the dividing ordinate *a d*, but unquestionably the vessels have shown the highest speeds when the value of *a d* has been the lowest, as far as the deep-bodied yachts are concerned. In the case of steam yachts, their straight length of middle body must be considered. In the table this has only been eliminated in the case of Capercailzie, as indicated by an asterisk.

Until within the last few years it was usual to find a great deal of hollow in the water-lines of the fore body at about the mid depth; but with the rounding up of the fore foot the fine parts of the water-lines have been removed, and without increasing the resistance at high speeds, whilst at low speeds the frictional resistance has been decreased by reason of the reduction made in the immersed surface (see pages 144, 145, and 168.)

The After Body.

Upon first examining the after bodies and water lines of the after bodies of various vessels, there would appear to be a great divergence in their forms, but if examined by their buttock lines, or ribband lines, there will in reality be found a striking similarity in them. The buttock lines closely resemble in character a parabola, as will be found after studying the table which here follows, which gives the factors for computing a parabola, and also for computing the mid buttock line of several well-known vessels.

Name of Yacht.	0	1		3	4	5	6	7	8	x
Parabola*	·000	·234	·437	·609	·750	·859	·937	·984	1·000	
Audrey (cutter), 1896	·000	·210	·442	·575	·708	·833	·896	·959	1·000	·12
Ghost (20-rater)	·000	·189	·364	·544	·694	·831	·920	·968	1·000	·20
Dolphin (2·5-rater)	·000	·230	·400	·540	·680	·820	·900	·964	1·000	·25
Minerva (21-rater)	·000	·187	·400	·583	·733	·857	·933	·983	1·000	·21
Volunteer (American cut.)	·000	·248	·466	·644	·778	·889	·954	·995	1·000	·16
Sappho (schooner)	·000	·225	·422	·580	·733	·844	·930	·982	1·000	·18
Aline (schooner)	·000	·240	·500	·675	·818	·909	·950	·981	1·000	·17
Florinda (yawl)	·000	·247	·476	·647	·790	·885	·951	·981	1·000	·20
Latona (yawl)	·000	·236	·472	·654	·790	·890	·954	·982	1·000	·22
Fiona (cutter)	·000	·225	·420	·600	·755	·860	·930	·982	1·000	·16
Sleuthhound (cutter)	·000	·200	·368	·547	·684	·810	·897	·958	1·000	·16
Kriemhilda (cutter)	·000	·210	·400	·575	·720	·840	·923	·980	1·000	·18
Vol-au-Vent (cutter)	·000	·195	·400	·552	·700	·825	·920	·972	1·000	·20
Vanessa (cutter)	·000	·223	·420	·575	·700	·820	·920	·980	1·000	·17
Egeria (schooner)	·000	·216	·425	·600	·750	·866	·941	·983	1·000	·22
Egeria, altered	·000	·200	·400	·575	·733	·850	·933	·980	1·000	·18
Freda (20-tons, cutter)	·000	·216	·397	·555	·683	·800	·900	·983	1·000	·18
Saraband (10-tons, cutter)	·000	·198	·375	·558	·719	·834	·913	·970	1·000	·17
Neptune (10-tons, cutter)	·000	·220	·400	·580	·730	·860	·930	·980	1·000	·18
Lenore (yawl)	·000	·232	·438	·607	·753	·866	·945	·982	1·000	·18
Genesta (cutter)	·000	·200	·375	·550	·722	·847	·930	·986	1·000	·19
Constance (yawl)	·000	·200	·408	·595	·708	·829	·929	·985	1·000	·20
Miranda (schooner)	·000	·233	·449	·614	·770	·877	·947	·989	1·000	·21
Oriental (S.Y.)	·000	·273	·523	·712	·846	·936	·970	·992	1·000	·23
Amazon (S.Y.)	·000	·220	·425	·600	·750	·875	·955	·990	1·000	·22
Fair Geraldine (S.Y.)	·000	·250	·475	·675	·787	·875	·937	·972	1·000	·19
Capercailzie (S.Y.)	·000	·375	·633	·808	·883	·941	·990	·998	1·000	·21
Capercailzie (S.Y.)†	·000	·300	·550	·710	·830	·900	·935	·975	1·000	·24
Chazalie (S.Y.)	·000	·260	·512	·700	·810	·900	·940	·970	1·000	·27
Marchesa (S.Y.)	·000	·350	·622	·800	·917	·970	·980	·985	1·000	·27

* The factors in this line are for computing a parabola.

† This is with No. 8 ordinate taken 12ft. abaft the greatest section to allow for the 24ft. straight of breadth.

The factors are thus found. A buttock line is put in the body plan midway between its middle vertical line and the greatest section; this buttock line is then projected on the sheer drawing in the usual way; and the *distance* from this section to the point where the buttock line cuts the load water-line near the stern post is taken as the *length* of the line, except in the case of the raking midship section. This length is divided into *eight equal* parts, and an ordinate is then drawn at each division perpendicular to the load water-line; the lengths of these ordinates are measured and consecutively *divided by the lowest ordinate,* which will be obtained at the midship section; the quotients are fractions, and are the same as the factors in the table. Of course if the midship section is raked the length of the buttock will be increased, and the greatest ordinate will be further forward. The fractions under x in the ~~table~~ is the distance in terms of the length of the after body that the buttock line cuts the load line ahead of the after end of the load line.

It will be seen that these curves closely assimilate to a parabola, but there are certain notable departures.

It could not be pretended, so far as we know, that there is any special advantage in an exact parabolic curve for a buttock line; but, undoubtedly, such a curve is fairer and truer than the curves of buttock lines occasionally met with.

At first it might be thought, looking at the buttock lines of some yachts which have comparatively short after bodies, that they have buttock lines of unusual curvature; but this may or may not be the case. The factors for the lengths of the ordinates need not necessarily be greater than those for a parabola to produce apparently full buttock lines, as the apparent fulness could be obtained by placing the ordinates closer together, as indeed they naturally would be in a very short after body.

The Genesta affords an example of a full-looking buttock line in a narrow vessel, and other examples can be found among the plates. In the deep and beamy type of boat it is difficult to get long buttock lines without giving considerable rake to the midship section, and when this is done it is usual to put the greatest breadth of the load water line farther aft than it otherwise would be; and often the practice is followed in narrow vessels. The Minerva and Isolde are extreme examples of what is done in this way in broad and narrow boats.*

* The "raking" or diagonal midship section has for a great many years been favourably looked upon by many naval architects. The "rake" can be thus described: Assume that the greatest breadth of the load water-line is 10ft. abaft the centre of length of that line; then that the greatest breadth of the lower water-lines is consecutively 1ft. (or more or less) forward of each other's greatest breadths; a line through points fixed at the various greatest breadths will determine the rake of the midship section. The original "Una" boats had it in a very

Formerly a great deal of contrary flexure was observable in the buttock lines near the sternpost (see the lines of Latona among the Plates). This was owing to the load line generally terminating some distance inside the stern post, and to the breadth and length of the counter. By extending the rabbet for the planking to the aft edge of the sternpost and much narrowing the counter, the buttock lines in most cases terminate without showing any hollow, and add much to the appearance of the yacht.

In the table the factors for computing the form of the mid-buttock line of several steam yachts are given. Most of these yachts have long flat floors or great length of straight middle body abaft the midship section; hence to compare them with sailing yachts, this piece of middle body should be left out of the curve. The effect of doing this has been shown in the table in the case of Capercailzie. In steam yachts driven by the screw propeller the after body requires the greatest attention in designing, and will be referred to further on.

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marked degree, and they were commonly described as "club-footed." But long before the advent of the Una boats Mr. T. D. Ditchburn, in 1835, built a vessel with a very great rake to the line of greatest breadths, and she was converted into a yacht because she would not carry sufficient cargo to be commercially successful. In 1860 Mr. Joseph Maudslay, the distinguished engineer, read a paper at the Institution of Naval Architects on this particular form of vessel, and the paper was accompanied by a drawing which exhibited very beautiful lines. The following is an abstract from Mr. Maudslay's paper:

"In designing a vessel of any given displacement, so as to obtain the greatest amount of speed, the water-lines at the entrance and delivery should be made as long and as easy as possible; and it appears that these lines may be made much longer and finer than they are in vessels as at present built, by placing the greatest breadth on the load-water line considerably abaft the centre of length of the vessel, and the greatest breadth on the lower water lines to the same extent forward of the centre of length, thus making the line of the cross section at the greatest breadths incline backwards at an angle from the keel, instead of its being in a vertical line, or at right angles to the line of the keel. This angle will necessarily vary in different classes of vessels, but may be taken at about 30° to 35° for vessels of ordinary proportion as to breadth and depth. It will be seen at once that this modification of the water lines offers great advantages for attaining high speed, as the horizontal water lines at the entrance will be much finer, and the fulness of these lines aft will be compensated by the increased fineness of the buttock lines, so that the water will be more easily divided and thrown aside forwards, while the void astern caused by the progress of the vessel will be filled by the water flowing from below in the direction of the buttock lines. By these means an increase of speed would be obtained in any vessel of a given amount of beam, without at all interfering with the amount of displacement; or an increased amount of stability might be given to a vessel, so as to enable her to carry more sail, by adding to her beam without at all adding to the resistance of her hull through the water."

It is quite a debateable point whether or not a vessel can be made to carry more sail by such a conformation; and it is equally open to doubt, if the centre of gravity of the plane of flotation were very far abaft the centre of gravity of displacement, if the vessel would be so easy in a sea as anticipated. At the time when Mr. Maudslay's paper was read, a discussion on the system took place, and the conviction seemed to be that it had advantages. Mr. Scott Russell declared that by raking the line of greatest breadths an entrance could be obtained which would be much longer than the half length of the vessel; and a run could be obtained which would be much longer than half the vessel. At any rate, it is a convenient method for obtaining good-looking buttock lines.

Form and Area of the Midship Section.

With regard to the form of the midship section, the manner it influences stability and resistance has already been explained, and in some respects these influences are antagonistic. For instance, it will be seen upon reference to page 43 that deepening the bilge, or carrying the greatest breadth well below the water line, and causing it to turn in rather sharply, as in the case of the 20-tonner Freda (see her mid-section, page 156), greatly adds to stability; on the other hand, upon reference to page 145 it will be observed how detrimental a deep bilge must be for speed on account of its aggravating the tendency to make deep transverse waves. It is therefore obvious that in adding to the displacement by deepening the bilge to augment stability that its effect on resistance should be well considered.

If a small weight of ballast is to be carried, a high bilge and hollow garboards will be resorted to, and flatter looking buttock lines will be obtained, extreme cases being found in the Dolphin, Kismet, Sappho, &c. There will, however, be an increase in the immersed surface in a case like the Sappho's, and this may be a serious matter for sailing in light winds. The full midship section with faint bilge, such as Genesta, Ghost, and many others have, possess the advantage of yielding a relatively small immersed surface. With such sections, however, the transverse wave-making will also be great, and the very highest speeds will not be attainable for any given length; still, it has been found in practice an advantage to have mid-sections approaching the form of those under consideration, the Minerva's being a good example; and it should be borne in mind that the results of competitive sailing are rarely determined by having a form adapted for the highest possible speed.

Such a section as Genesta's is not, however, practicable, where speed is concerned, in vessels of great beam, unless the depth of under-water body is very limited.

It is usual to compare the areas of midship section as a measure of resistance, but, as exemplified on page 138, the comparison except in vessels of exactly similar form may be entirely misleading. The general character of the effect of the area on speed will be gathered from the chapter on "Resistance," particularly on pages 145 and 155.

In yachts designed in accordance with the wave form requirements the area of the mid-section is about 1·75 times the area of the average area of

the sections. A formula for approximating the area of mid-section in such vessels would therefore be

$$\text{Area mid-section} = \frac{\text{Displacement}}{\text{Length L.W.L.}} \times 1{\cdot}75$$

the displacement being expressed in cubic feet.

Or the displacement could be found from

cu. feet

$$\text{Displacement} = \frac{\text{mid-section}}{1{\cdot}75} \times \text{L.W.L.}$$

In vessels finer, either in the fore body or after body, than the wave form theory prescribes, the proportion of mid area to the average area of section is larger; if the vessel is fuller than a wave form the proportion is smaller, the range being from 1·6 to 2·2 in examples we have examined.

CHAPTER XI.
STEAM YACHTS.

THE BOILER AND ENGINE.

In spite of the number of Institutions, both in this country and abroad, devoted to the record and investigation of physical science; and in spite of the many trials and experiments made from time to time by our own and foreign naval authorities, the fact remains that there is no source from which a comprehensive record of the progress of modern marine engineering can be compiled. The transactions of the Institution of Naval Architects and other scientific societies, and the pages of our principal engineering journals, no doubt record the leading features in the march of improvement, and they are also rich in information on numberless points of detail as to marine engineering practice; but when we go further, and inquire as to performance and efficiency, we find no complete record which forms a continuous and unbroken history of marine engineering.

What is principally wanted in the present day is a comprehensive series of experiments on a large scale, conducted by competent and unprejudiced engineers, in order to determine the relative efficiencies of marine engines and boilers of various types. We want done for the marine engine what Mr. Froude accomplished in the field of naval achitecture by means of model experiments, and trials with the Greyhound. The engineering question, however, is far more complex than that with which the naval constructor has to deal, and model experiments are not admissible in the former case as they are in the latter.

We do not mention these facts merely for the sake of pointing out a difficulty, and in order to form an excuse for the necessarily imperfect nature of this chapter, but in the hope that some owners of the many fine vessels in the English yachting fleets will utilise the exceptional opportunities at their command, and give to marine engineering science some of that practical information it now so sadly lacks.*

* Since this was written the Institution of Mechanical Engineers has appointed a research committee to make trials and report on steamship machinery. Up to the time of publication the labours of the committee have not produced results which render the above passages no longer true. The committee has, nevertheless, done some good work, and promises to do better. Further on we quote from the reports already presented.

In considering the subject of steamship machinery, the steam generating apparatus claims our first attention, both in natural sequence and in order of importance. It is on the boiler that the efficiency, safety, and comfort of the whole machine mainly depends. No vessel with a boiler too small or ill-designed can do good work, whilst we often see steam craft of all kinds which manage to get along fairly well with very questionable èngines. There is one axiom that all yachtsmen should take to heart: a small boiler, *i.e.*, small for the work it has to do, is always a bad boiler. The boiler, to those who are not engineers, is not a very interesting subject, and too often both owner, designer, and constructor combine to cut down the space allotted to this important element. The mistake is invariably fatal, for a dwarfed overworked boiler always revenges itself—too often at the most critical times.

A perfectly efficient steam generator would be one that would give in steam an equivalent to the full theoretical value of the fuel burnt, and such a boiler naturally only belongs to the domain of theory. A pound of good Welsh coal is theoretically capable of evaporating at atmospheric pressure, about 15lb. of water, from a temperature of 212°. This duty, as we have intimated, is not reached in practice for a variety of reasons, the chief of which are the waste heat which escapes by the funnel and the loss due to the gases evolved from the fuel being carried to the chimney without being burnt. By bad stoking or defective grate bars a good deal of fuel is often wasted, and there is a certain amount of heat lost by radiation, especially if the boiler be not covered with some non-conducting material, such as felt, slag-wool, &c. The loss which arises from the escape of heat by the chimney is not to be avoided, as the draught necessary for urging the fires is due to the lighter and hotter gases ascending in the funnel in all boilers run by natural draught. From this the advantages offered by a high funnel will easily be seen, when looked at from a utilitarian point of view, however much the susceptibilities of those who mix up æstheticism with steam boiler chimneys may be offended. The best temperature for the products of combustion to escape at is generally considered to be about 600° Fahr. With forced draughts, such as is used in torpedo boats, there need be no difference between the temperature of the chimney gases and the outside atmosphere, so far as draught is concerned; but naturally the products of combustion can never be brought lower than the heat of the water and steam in the boiler. As a matter of practice they are never nearly so low; the large additional amount of heating surface required, when the equilibrium in temperature between the two has been at all closely approached, would be so great as to render the boiler costly and

heavy out of all proportion to the saving in fuel. By means of feed water heaters, the waste heat escaping from the uptake may be utilised. The heat is transmitted to the much colder water of the feed far more readily than to the hot water in the boiler. A thoroughly satisfactory appliance of this kind would afford a great saving of fuel, especially if it were worked in conjunction with the forced draught from a fan blower.

From what has been said it will be seen that one of the chief causes of loss referred to is inevitable with our present system of steam generation, and the marine engineer in designing a boiler does not aim at lessening the heat of the waste products, as he would thereby destroy his draught. The case is, however, different with the second source of loss mentioned. Every atom of unconsumed gas that escapes at the chimney top is a complete waste of fuel; and not only is it a direct loss in itself, but it carries with it the heat which has been absorbed in transforming the fuel from a solid to a gaseous state. The size of flues and combustion chambers bears especially on this question, and these we refer to again later on. It is necessary also that a sufficient amount of air be admitted to the furnace, in order to supply the oxygen required for perfect combustion; perfect combustion being the chemical combination of the furnace gases with the oxygen. No heat is given off by the distillation of the gases from the fuel; in fact, the reverse is the case, for heat is absorbed. Unless the atoms of gas are brought into contact with the atoms of oxygen, both being at the requisite temperature, it would be more profitable to throw the coal from which such gases are evolved into the sea, rather than put it on to the grate. About 12lb. of air are theoretically necessary for burning each pound of coal; but as theoretical considerations by no means govern the question, it is found in practice that from 18lb. to 25lb. of air must be supplied to insure the combustion of each pound of coal; the considerable difference in quantity being accounted for by the description of coal that may be used, and the briskness of draught. With hard coal, in which the percentage of carbon is high, when burnt by forced draught, less air is required to pass through the furnace than with bituminous coal and sluggish draught.

When the air passes up through the grate bars the oxygen in it combines with the carbon and forms carbonic acid gas. This should be the product of combustion discharged from the funnel, as in that case the carbon and oxygen have been brought together in suitable proportions, and all the heat that it is possible to get has been obtained from the carbon in the fuel. Carbonic acid gas has, however, the property of taking up a further quantity of carbon, and carbonic oxide is thus formed. The excess of carbon in the latter can be burnt by being brought in

contact (always at a sufficiently high temperature) with a further quantity of oxygen, and carbonic acid gas is again formed. This of course is incombustible, and incapable of being used for heating purposes. It will be seen from this how necessary it is, especially with thick fires, to provide for admission of air above the furnace bars. When using bituminous coal there is additional necessity for a supply of air other than that which passes through the furnace bars. The hydrogen set free from such coal requires a larger supply of oxygen to effect the complete combustion of any given volume than does carbon, and should such oxygen not be present, the hydrogen will be wasted, generally escaping at the funnel in combination with a greater or less degree of carbon, either as marsh gas or olefiant gas. On the other hand, too much air will carry off the heat to waste, and, the gases being diluted and so cooler, the rate of transmission will be less, thus making the heating surface less effective. Moreover, the greater the volume of the gases and products of combustion the more quickly they will pass through the tubes, and therefore the less time will they have to part with their heat. Supposing the temperature of the atmosphere to be 60°, if the fuel could be burned with 12lb. of air to each pound of coal the the temperature of the products of combustion would be 4640°. If 18lb. of air be used the theoretical temperature will be 3275°, and with 24lb of air per pound of fuel the theoretical temperature will be 2500°.

It is difficult to give any exact rule as to the total area of opening by which air may be admitted above the fuel. The holes required should not be too large, about ½in. in diameter perhaps being a maximum. The total area of opening may range between 2½in. and 5in. per square foot of bar surface, but the conditions vary according to the description of fuel used, the speed of draught, and other considerations. A greater volume of air above the bars is required immediately after firing, as it is then the hydro-carbons are chiefly liberated, and these, as we have intimated, require a larger volume of air to complete their combustion than the carbon, which is liberated more slowly.

The principal class of boiler in modern sea-going vessels, such as yachts, is the cylindrical return tube type, with either one, two, three, or four furnaces. Although somewhat costly to manufacture, it possesses advantages which have enabled it to hold its own as the best in marine practice, the chief of these being its reliable character, as tested by years of experience, combined with fuel economy, and the small length it occupies in the vessel. As will be seen, other types of boiler are coming forward which bid fair to contest its supremacy. Return tube boilers are divided into two classes, single-ended and double-ended. The latter are fired at each end, and the combustion chamber, or chambers, are situated

in the middle of the boiler, the tubes returning over their respective flues in the usual way. The positions of furnaces and combustion chambers may be arranged in either of the following ways. There may be a single combustion chamber in which all the furnaces terminate; there may be two combustion chambers placed back to back, so that the furnaces in each end of the boiler have separate chambers; there may be a combustion chamber common to each pair of furnaces opposite each other; or there may be a separate combustion chamber to each furnace. The latter plan has many advantages, but it is costly to construct, and makes a heavy boiler. Double-ended boilers are, however, seldom necessary in the case of yachts, and we may therefore confine our attention to the second type mentioned: namely, the single-ended boiler. This may have either one, two, or three furnaces; but there are comparatively few yachts large enough to require three-furnace boilers. Each plan has its particular advantages and disadvantages, and there is much diversity of opinion as to the best practice in this respect. The balance, however, is decidedly in favour of a single furnace whenever it can be brought within manageable limits. One of the principal requirements is to give sufficient height above the grate bars, in which the gases may be burnt in the manner already described. It will be readily understood that what is known as the heating surface of the boiler is heating surface only so far as the water is concerned, being, on the other hand, cooling surface to the gases. If, therefore, the gases are brought in contact with the sides of the furnace before they have combined with the necessary oxygen they are cooled below the temperature of combustion; and, unless they are again brought to the required heat (which may or may not occur), go to waste up the chimney. It will be easily seen how desirable, therefore, it is to have a fair amount of space above the grate bars, and this space is greatest with large diameter furnaces. It may be mentioned in illustration of this part of our subject, that the old-fashioned rectangular box boilers, with furnaces of square section, were, as a rule, more economical than the modern cylindrical type. A large combustion chamber, although an excellent feature, will not altogether compensate for a furnace of small diameter, as the gases may be reheated whilst they are exposed to the radiant heat of the furnace, but the chance is very much diminished when they get into the combustion chamber. The single furnace boiler is also cheaper to make, is lighter, and further possesses the advantage of giving less space for dead water at the bottom; a detail that considerably affects the life of the boiler, especially in the case of vessels, such as yachts, in which the steaming is intermittent.

The points that tell against single furnaces are that a longer grate is

required to get a like area, and the steam will fall in pressure during stoking. By stoking two furnaces at different times the steam pressure can be kept more regular, and when there is a combustion chamber to each furnace the whole boiler is not incapacitated by a defect in a tube. The decision as to the number of furnaces must of course depend on a variety of circumstances, such as the amount of grate surface required, the pressure at which the boiler is to work, and other considerations of this nature. As a general rule in ordinary practice, 4ft. may be taken as an extreme diameter, whilst there are few furnaces below 2ft. 4in. Large marine boilers are made with four furnaces. Double ended boilers, such as the steam yachts Mohican and Fire Fay are fitted with, four furnaces—two in each end. Speaking generally, and taking ordinary pressures, return tube boilers up to 7ft. 6in. diameter, would be made with a single furnace; up to about 13ft. 6in. with two furnaces, and above that diameter with three furnaces. The Imogen, built at Paisley in 1890, has a boiler 14ft. 6in. in diameter with three furnaces.

One of the greatest improvements in the modern high pressure boiler has been the invention of the corrugated flue (see Fig. 97), the intro-

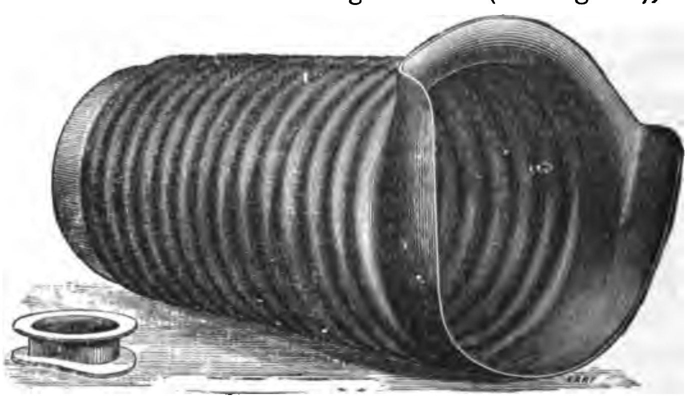


Fig. 97.

duction of which is due to Mr. Sampson Fox, of the Leeds forge. These are rolled solid by special machinery of great power, and there are no seams of rivetting in the place where of all others they are more likely to give trouble. Of the value of this invention it is unnecessary to speak, as it has received the highest testimonial possible in the shape of general application in the best marine practice. When first introduced, several of these furnaces gave out, but this resulted from their being made of iron, the lamination of which caused the trouble. Mild steel only is now used, with the most complete success. The advantages in this class of furnace are, that greater strength is obtained with a thinner plate, and compensation ia allowed for the stress due to unequal expansion and contraction.

The heating surface in the furnace itself is increased, and probably a good effect is produced by the deflection of the gases from the surface of the furnaces, and their consequent more thorough mixing with the air. The Purves furnace is in principle similar to the corrugated furnace, and has come extensively into use of late years. There are also one or two other furnaces of a like nature, but differing in detail of construction.

The tubes in a boiler are generally made of iron or steel in mercantile vessels. Formerly ships of the Royal Navy had brass tubes, but steel has now made its way in the Service too.

For some time the preference appeared to be becoming more general for tubes of smaller diameter, but, tempting as it is to get a large amount of heating surface in a small space, the practice may easily be overdone. The general rule is to make the diameter of the tubes one-twenty-fourth of their length if a fair height of funnel can be obtained. Mr. Milton, of Lloyd's, has given some instances of the effect of reducing the size of tubes. Taking $3\frac{1}{2}$in. as a standard, and maintaining $1\frac{1}{4}$in. spaces for cleaning purposes, as is usual in good practice, we find that with $2\frac{3}{4}$in. tubes 10 per cent. extra, and with 2in. tubes 20 per cent. extra surface is obtained; while if the space were reduced to $1\frac{1}{8}$in. with the $2\frac{3}{4}$in. tubes, and 1in. with the 2in. tubes, which would probably be found to be equally efficient for cleaning when we consider the reduced size of the tubes, these figures are increased 18 per cent. and 43 per cent. respectively. The total area of the tubes in cross section should not be less than one-seventh the grate area in ordinary practice with moderately high funnel. If the tubes are placed too close to each other priming is likely to occur. Too small tubes tend to throttle the draught, causing slower or imperfect combustion in the furnace; if, however, the tubes afford too large a total area, the draught is not sufficiently brisk in the tubes themselves, and they are more liable to get choked. The smoke box into which the tubes discharge should have a sufficient lengthways depth, so as not to check the issue from the tubes; the depth should be from 12 to 15 inches, and the uptake from the smoke box should lead direct to the funnel.

It is generally considered that the combustion chamber in a return tube boiler should be equal in capacity to the furnace in order to give good results with boilers designed for ordinary pressures. Shortcomings in this respect will cause a low efficiency. This point, however, is better understood in the present day than it was a few years ago, the average depth of combustion chambers having been almost doubled of late years in best marine practice. It should be noted that quite recently a large number of very big return tube boilers have been constructed to work at high pressures, with furnaces very large in diameter and the combustion

chambers comparatively small. No doubt this is good practice, as manufacturers can now supply furnaces that will stand. The part that makes a return tube boiler so costly is the combustion chamber, but this additional expense may be avoided by the use of a dry combustion chamber. In this type the boiler is a simple cylindrical shell, the furnace extending right through, and the tubes returning from the back tube plate in the usual way. In place of the water jacket to the combustion chamber, with its flat surfaces and numerous stays, a casing of fire brick is built up at the back end, forming a passage for the heated gases between the furnace and the tubes, in much the same way as the uptake over the furnace mouth makes a communication between the tubes and the chimney. By this arrangement it will be seen that a great deal of very effective heating surface is lost, its place being taken by fire brick. The loss, however, is more apparent than real, for the tube surface is rendered far more effective. Probably in ordinary diameter tubes combustion of the gases does not take place for a further distance than three or four inches from the end of the tubes, as the gases are speedily brought below the necessary temperature by the heat transmitted from them to the water surrounding the tubes. It will therefore be seen that unless the gases are burnt in the furnace or combustion chamber they are likely to get away through the tubes unconsumed. It will also be understood from what has been said that a surrounding envelope of water is likely to check combustion by the abstraction of heat. Fire brick being a very bad conductor of heat the gases do not become lowered in temperature, and the combustion chamber is more likely to play its proper part.

The extent of grate surface is the governing principle in designing a cylindrical boiler, as the width of grate determines the diameter of the furnace, and also the length of the grate. Long grates are difficult to stoke, and in any case the length should not exceed twice the diameter of the furnace, but to get the best results about one and a half diameters will be sufficient, although a great deal depends on the skill of the fireman. Fire bars may be in one, two, or more lengths. The Admiralty prefers not more than 2ft. 3in., but bars 5ft. 6in. are worked satisfactorily in ordinary practice. It is better to have the bars in one length if possible, as the bearer, which interferes a good deal with the cleaning of the furnaces, is then avoided. Thin deep fire bars are preferable, as they are less likely to become overheated.

The relative proportions of grate and heating surface most suitable in a return tube boiler vary with the conditions under which it is to be worked, the quickness of draught and description of fuel being the most important considerations. In a return tube boiler of good design 1lb. of coal will

evaporate, under favourable conditions, 9lb. to 10lb. of water, but in ordinary work 8lb. is a fair result. In yachts running with natural draught, from 14lb. to 20lb. of coal can be burnt per square foot of grate surface per hour; but the larger quantity named is seldom consumed, except on trial trips, when great attention is paid to the stoking and draught. Taking the larger quantity, as representing what is done on trial trips, and allowing 2lb. of coal per I.H.P., we should require 0·1 square foot of grate surface per I.H.P. With iron tubes from 2 to 3 square feet of total heating surface is generally allowed for each indicated horse power required; but the rule is by no means uniform for yacht practice, as will be gathered from the table, and it will be noted that the efficiency of the grate diminishes very rapidly as its area decreases in small vessels.

	Tighnamara (late Imogen).*	Fleur de Lys (late Malikah).	Fauvette.†	Mazeppa.	Oriental.	Primrose.	Celia.	Linotte.	Marchesa (late Queen Marfisa).
I.H.P. on trial trip	1071	441	1150	375	330	101	58	197	246
Heating surface	sq. ft. 1800	sq. ft. 1250	sq. ft. 2300	sq. ft. 1057	sq. ft. 696	sq. ft. 333	sq. ft. 206	sq. ft. 676	730
Grate area	63	50	84	40	30	12·3	8·7	25	30
Heating surface to grate area	28·6	25	27·4	26·4	23·2	27·1	23·6	27	24·3
Square feet of heating surface per I.H.P.	1·671	2·834	2·0	2·819	2·109	3·297	3·378	3·432	2·967
Square feet of grate area per I.H.P.	0·588	0·113	0·76	0·107	0·091	0·122	0·150	0·127	0·122
I.H.P. per square foot of grate area	17*	8·8	13·7	9·4	11·0	8·2	6·7	7·9	8·2
Boiler pressure	160lb.	150lb.	160lb.	80lb.	80lb.	80lb.	75lb.	100lb.	100lb.

* The indicated horse power for Imogen is given on the authority of the builders as being obtained on the trial trip with induced draught caused by a steam jet blast in the funnel, a practice not admissible for ordinary running. 17 I.H.P. per square foot of grate is nearly double that obtainable with natural draught. 10 I.H.P. per square foot is regarded as a very fair achievement even for a trial trip, and 8 I.H.P. for ordinary running with natural draught.

† Fauvette for the I.H.P. given used Martin's "induced draught."

An important consideration in designing a marine boiler is the steam space that should be allowed, for on this mainly depends whether the boiler will prime or not. A rough general rule is to keep the top row of tubes one-third the diameter of the shell from the top, whilst some engineers calculate the contents of the space above the water level, allowing from half to three-quarters of a foot per I.H.P. It is, however, not only the actual capacity of steam space, but to a large extent the area of water plane in a boiler that affects priming. What is required is to draw the steam equitably from as large a surface as possible, and for this reason an internal pipe with saw cuts on its upper surface is placed under the top of the boiler shell.

With the increase of pressures the steam space has been reduced in marine boilers, for the reason that the higher the pressure the smaller the

volume for any given weight of steam, the difference in volume between steam of 60lb. and 100lb. pressure being approximately as follows: 6·8 cubic feet per pound of steam at 60lb. pressure, and 4·3 cubic feet at 100lb. Where the required steam space cannot be obtained without much increasing the diameter of the boiler, a steam dome is fitted, an example of which is given on Plate VI. It is generally considered that when steam is taken from a dome the chances of priming are reduced.

In the present day steel may practically be said to have superseded iron as a material for boilers of the best class. The life of a yacht's iron boiler of 75lb. pressure has ranged from ten to twenty years, according to the use and treatment it has been subject to; and, so far as experience has yet gone, there is no reason to suppose that steel boilers with pressures ranging from 100lb. to 180lb. will be less durable. But the latter require much more careful treatment than the old iron boiler of 70lb. or 80lb. pressure, or there will be trouble with the tubes and tube plates or furnace crowns. Steam must be got up very slowly in from four to five hours, and care taken that there is an early circulation; also the boiler should be cooled down with equal care by closing the damper (in the funnel), the ashpits, &c., and not by drawing the fires. Particular care appears to be necessary in this respect on first using a boiler, and some engineers will not allow a new boiler to steam to its full pressure on the first time of using it. It is also necessary that the boiler should be fed with fresh water to make up for waste. Mr. Parker, chief engineer to Lloyd's, in dealing with this subject in 1890, said:

It is known that deposit takes place more rapidly with high pressures, and consequently high temperatures, than with low-pressures. Theoretically speaking, by the process of converting water into steam, condensing it into water, and again converting it into steam, no loss whatever should take place; but in practice there is a considerable loss from leaky pumps, cylinder glands, cocks, safety valves, condenser tubes, &c , and in an engine kept in good working order, all parts being, practically speaking, tight, and no loss from steam blowing off, &c., I find that for every thousand horse-power exerted, one ton of water per day has to be added to the boiler in the form of extra supply. If this water has to be taken from the sea, no less than 2 cwt. of solid matter must be deposited every day, and it can easily be seen how detrimental such treatment must be to boilers engaged on long voyages. It is considered that all boilers working at high pressures should have some means of making up this loss in the form of fresh water, either by carrying it in the vessel, or evaporating and condensing.

In working marine boilers at the high pressures now so common, a great amount of ignorance is often displayed by the engineers to whose care they are entrusted, and it may be stated that nearly all the cases of collapsed furnaces, leaky tubes, &c., that have come under the notice of the officers of this society since the introduction of the high pressures, have been traceable to improper treatment.

Rules for constructing marine boilers are laid down in various text books, and the two sectional views of a modern return tube boiler (Plate VI.) will serve to give a general idea of a well designed boiler, but it will of course be understood that the mountings are not shown, the

furnace doors, uptake, &c., being left out. The end plates are flanged both for forming the connection with the shell plating and with the furnace, according to the most approved practice. The boiler was constructed, in accordance with Lloyds' rules, in 1888 by Messrs W. White and Sons for the steam yacht Linotte from designs and instructions furnished by the owner, M. Perignon, the eminent French engineer. Another good boiler fitted to the steam yacht Speedy, built in 1895 by Messrs Ramage and Ferguson for the Baron Barreto; and that of the Amazon, and made by Messrs. Day, Summers, and Co., is illustrated on Plates VII. and VIII.

Amongst the boiler mountings the most important is the safety valve, ample rules for which are laid down by the Board of Trade and Lloyd's Society. The former requires that there shall be half a square inch of valve area to each square foot of grate area, and unless the grate area be less than 14 square feet there must be two safety valves to each boiler. Ships in the Royal Navy have a small weighted valve loaded to a few pounds above the working pressure. This acts as a warning in case the main safety valves stick. Steam gauges are on the Bourdon principle. They should be placed where readily seen. Check or non-return valves will of course be placed on all feed branches, both from main feed and donkey pumps. Water gauge glass and mountings, and try cocks, blow-off cock, scum cock, steam whistle, &c., all require attention, but the limits of our space do not permit us to go into details here. For clothing boilers hair felt is very commonly used, but it is likely to rot if it gets wet, or take fire when dry. Silicate cotton, which is made of blast furnace slag, blown, when in a molten state, by a jet of steam into fine thread-like fibres, is an excellent non-conductor, proof against rot and inflammability. It is apt, however, to crumble in course of time, and will ultimately become a fine dust. Asbestos fibre is probably the best lagging for boilers, but is somewhat costly. A newly-introduced substance called fossil meal has also been well spoken of.

Hitherto we have only treated of the return-tube boiler, which is all but universal in English sea-going vessels. Several of the smaller class of ships in the Royal Navy, in which it is necessary to keep the boiler below the water line, have been fitted with what is known as the gunboat boiler. This consists of a cylindrical outer shell, having two furnaces. The latter terminate in a common combustion chamber, on the further side of which is the tube plate; the tubes, in place of being returned above the furnace, are thus carried straight on to the back end of the boiler, where an uptake is fitted, into which they deliver. There is generally a hanging fire brick bridge in the centre of the combustion chamber, in order to distribute the

heat more equally through the tubes. Excellent evaporative results are obtained with this type of boiler. The locomotive type has been modified to suit marine practice, and was brought prominently into notice by Thornycroft and Yarrow in their fast steel launches. Such boilers are always made of steel, and their great advantage is their lightness. Mr. Alfred Holt has stated that a ton weight of locomotive boiler, will produce as much steam as 6 tons of ordinary marine boiler; but, it must be remembered, the conditions of running are dissimilar. For some time it was hoped by engineers, in view of the wonderful performances of torpedo boats, that the locomotive boiler, worked in conjunction with forced draught, would bring about that revolution in steam generation so long expected. The experience of the torpedo ram Polyphemus, however, indicated that the problem was more difficult than was at first supposed; and, indeed, few are aware of the troubles that attend the working of a first-class torpedo boat when pressed to her highest speed; at which point the data for comparison as to respective weights of boilers are generally taken. For small vessels in which a good artificial draught can be obtained, either by the exhaust steam in the chimney from non-condensing engines or by fan blast, the locomotive boiler is no doubt a good type if run with fresh water. The flat surfaces in the fire box are not difficult to stay excepting on the crown of the furnace. Unfortunately the evaporation here is very rapid, and with a fierce fire urged by a powerful blast there may be danger of the crown plates becoming overheated. Mr. Yarrow has introduced an ingenious arrangement of crown stays, by which compensation is allowed for expansion and contraction; in addition to this, the crown is raised near the tube plate. This plan keeps the top covered during motion in a heavy sea, and also allows for the expansion and contraction of the tubes. The latter is a serious difficulty with the locomotive type of boiler when pressed by powerful blast, the trouble commencing as soon as the pressure is allowed to fall or the fire is checked.

Both the gun boat boiler and the loco-marine boiler are, however, fast being superseded in the Royal Navy by the "Express" type of water tube boiler; indeed, at the time of writing (1896) it would seem that shell boilers of any kind are likely soon to be things of the past in a British war vessel. Not only are cruisers being fitted with water tube boilers, but line of battle ships are also to have them. At present there does not appear any great disposition for mercantile practice to follow this bold lead of the Admiralty engineering department. A few owners have been feeling their way in a tentative manner towards water tube boilers, but the results, up to the present, have not been altogether encouraging. No doubt some

of the failures that have taken place in mercantile ships have been due to untoward events not affecting the main design; but when this has been said the fact remains that the water tube boiler has yet to win acceptance at the hands of British shipowners.

Water tube boilers may be roughly divided into two classes—the express or quick steaming boiler, and the ordinary or large diameter tube boiler. The former are generally fitted into small craft, such as torpedo boats, whilst the latter are suitable for ships and large craft generally. The best known of the express boilers are the Thornycroft and the Yarrow types, so called after their respective inventors. As will be seen later, these two differ essentially in design; but it may be broadly said that all other types of express water tube boiler contain the leading features of either the Yarrow or the Thornycroft boilers. There are variations in details, and the changes are rung on the elementary conditions, but it is unnecessary to go outside the two types mentioned, at any rate, for illustrations of principles.

Among the larger boilers, the Belleville is decidedly the best known in this country, having been brought into prominence by its wholesale introduction in the British Navy. The Lagrafel D'Allest boiler, perhaps, stands next in prominence, it having been used largely in France. It is, in a way, a shell boiler also, the flat end chambers into which the tubes pass being exposed to the heat of the furnace. The large tube plates, although doubtless well stayed, appear to the writer as likely to be a source of weakness after a time; although it must be confessed that no serious trouble has arisen from this cause up to the present.

The express boilers are, perhaps, the most interesting to yachtsmen, and we will deal with them first.

The illustrations set forth overleaf in Figs. 98, 99, 100, and 101 are respectively a front elevation, a cross sectional elevation, and a longitudinal section of a Thornycroft boiler, of what is known as the Speedy type. The design will be seen to consist mainly of two horizontal cylinders at the bottom, and a larger horizontal cylinder at the top. These are connected firstly by a large number of bent tubes or pipes, as shown best in Fig. 99, and two larger straight tubes, as shown most plainly in Fig. 98. The grate is enclosed within dwarf fire brick walls at the sides, with a higher wall at the front end, and a still higher wall at the back end, the whole apparatus being contained within a light metal casing or smoke jacket. Such are the main features of the design; but, in order to understand the method of operation in the boiler, it is necessary to divide its functions into two parts, viz., those relating to the course of the smoke, furnace gases, and products of combustion in general.

We will deal with the water and steam first. Turning to Fig. 98 it will be seen by the position of the feed delivery valve that the feed water is pumped into the large upper cylinder, which may be called "the separator." Before lighting fires the boiler would be pumped up to its normal water level, or about a third way up the separator, as shown in Figs. 99 and 100. When water was first pumped in it would flow down the two big diagonal tubes at the end (these we will call the "down-comers"),

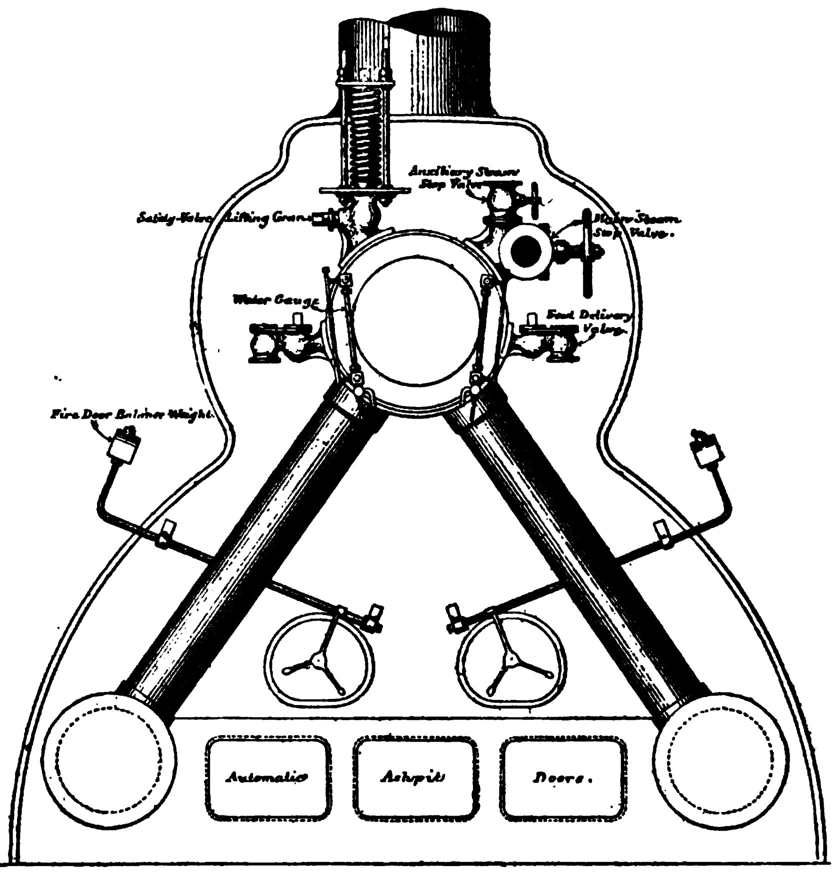


Fig. 98.

and into the two horizontal cylinders, which may be called the "wing cylinders." From the latter the water would flow into the small diameter bent tubes or pipes, to be called the "generating tubes." It will be seen, therefore, that when the water is at a given level in the separator—as indicated by the ordinary water-gauge glasses shown in Fig. 98—it will be at the same level in the generating tubes. As the top ends of the generating tubes open into the separator there is a clear right of way for

water (or water and steam) from the separator, down the down-comers, through the wing cylinders, up the generating tubes, and back into the separator again. This is the course actually taken by the feed water, as will be more fully explained presently. We must now trace the course of the products of combustion from the furnace to the chimney.

It will be seen by Fig. 99 that the generating tubes, by reason of the contour to which they are bent, inclose an arched space over the fire bars.

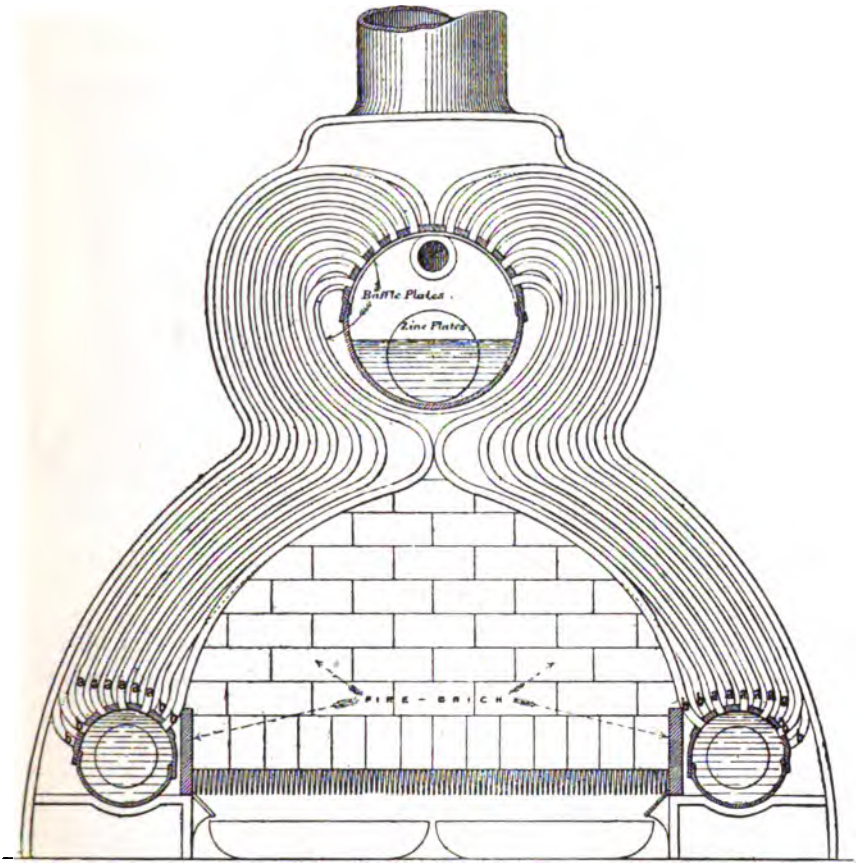


Fig. 99.

This arched space is closed at the ends by fire brick, and it constitutes the furnace. It is well to observe that the generating tubes, by curving towards each other at the crown of the arch, quite protect the separator from the radiant heat of the furnace. Heat having been evolved in the furnace, it must be conveyed amongst the generating tubes on its way to the chimney, part of the heat being, of course, absorbed by the inner part of the two rows of generating tubes which form the walls of the furnace.

The way in which these generating tubes are made to form a pair of flues is very ingenious. We may, for the purpose of explaining this

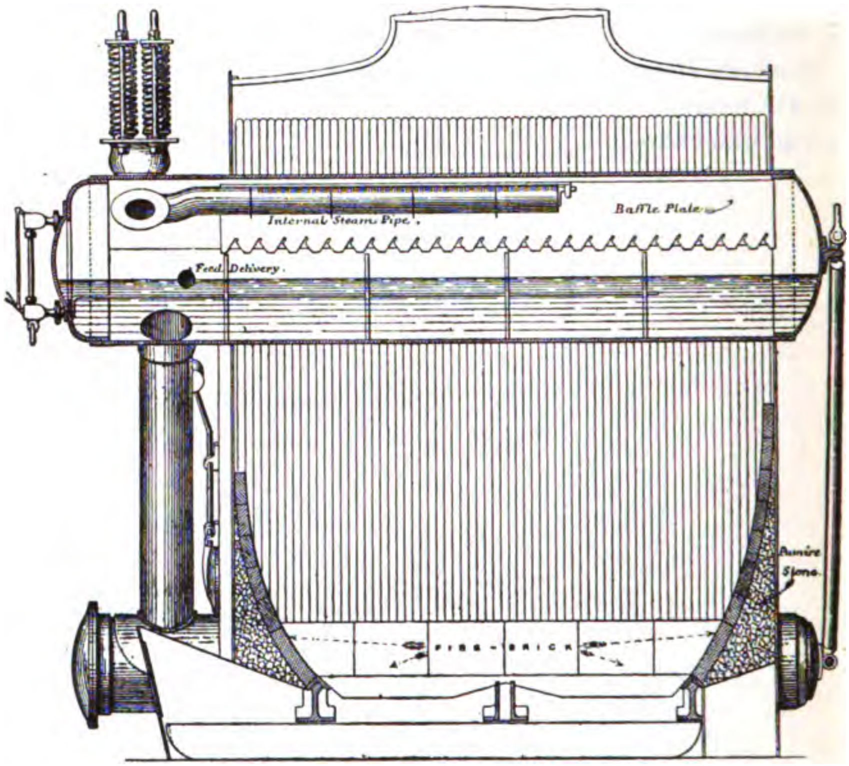


Fig. 100.

function of the generating tubes, suppose the middle series of tubes, *a a*, Fig. 99, removed, so that there only remains the outer row next the

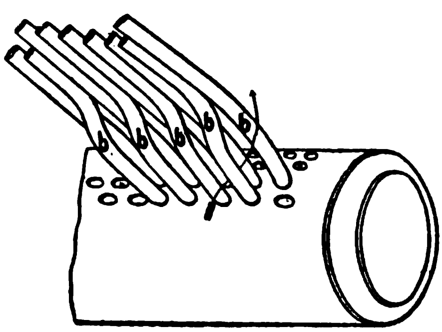


Fig. 101.

casing or smoke jacket, and the inner row next the furnace *b b* Figs. 99 and 101. These two rows form the walls of the flues which convey the

products of combustion to the chimney. It will be evident, therefore, that each tube must touch those on either side, otherwise the flue would not be smoke tight. It is necessary, however, that the products of combustion should pass from the furnace into the flue space, and this is effected through triangular openings at the bottom, where the generating tubes are let into the wing cylinders. Fig. 101 illustrates the arrangement. It will be seen that the inner row of generating tubes springs from two rows of holes, and that the tubes are bent together a few inches from their springing, so that they lie together with their axes in the same longitudinal plane. A little consideration will show that this is a necessity of the arrangement, for the tubes could not be placed side by side at the bottom (supposing they are to touch each other), as the holes into which they are expanded naturally require some spacing, as shown in Fig. 101. The tubes forming the outer wall of the flue must therefore have like triangular spaces. This would allow the escape of hot gases against the outer casing; but this is easily remedied by the application of a little fire clay. It will be seen that the wing cylinders are protected from contact with the burning coals by fire tiles or dwarf walls of fire brick.

Having got the heated gases and products of combustion into the flues, it will be easily understood how they ascend to the top and escape from the pipe-walled flue through other triangular openings, formed by the generating tubes being let into the separator in the same way as they are to the wing cylinders. The top part of the casing forms a suitable smoke box, or uptake, to which the chimney is attached. The inside series of tubes, *a a*, Fig. 99, are not bent at the ends, as are the outside ones, *b b* which form the walls of the flue. Each one, therefore, stands alone, not touching its neighbours, and the heated gases are allowed to play freely round them.

Having, we trust, made clear the arrangement so far, it remains to show how the circulation of water is obtained. This is a factor so essential to the successful working of a water-tube boiler that perhaps it will be well to first say a few words on the subject.

If we take the boiler of our ancestors, the simple iron pot with a closed top, we have a vessel for generating steam in which circulation is of the least consequence. The evaporation is slow, and natural convection, which takes place whenever water is heated, is not checked by stays or tubes. The bubbles of steam as they are formed can therefore easily force their way to the water surface. In a water tube boiler, with small diameter tubes of any considerable length, the conditions are entirely reversed. The water plane in each tube is exceedingly small, the generation of steam is remarkably rapid, owing to the small volume of water

acted upon, or rather to the excessive area of heating surface compared to the volume of water. The result is that when steam is formed in the lower part of the tube, supposing the tubes are arranged vertically, it has not time to pass through the water, and it therefore lifts the latter bodily, most likely forcing it right out of the tube. If a separator be not placed between the generating tube and the boiler the water will be driven into the engine cylinder. It was for this reason that the steam and water separators of various forms were introduced by the early designers of pipe boilers, whilst some endeavoured to reach the same end by throttling the steam. But these inventors did not sufficiently appreciate that there was another result which would follow the lifting of the water, hardly, if any, less harmful than a cylinder full of water. When the generation of steam is so rapid, and the water space is so restricted that the steam cannot rise through it readily, the water gets driven to the upper part of the tube, being supported by the cushion of steam below. This steam becomes superheated, and so much so that, in extreme cases, the tube might become red hot. Of course, the heavier water above is always trying to get to the lowest place in the tube, just as the lighter steam attempts to get uppermost; but this reversion to a more natural state of affairs is prevented by the particles of water as they descend in the tube becoming evaporated almost instantaneously, and before they can get to the lower part of the tube.

It is just this tendency of pipe boilers to prime with violence that Mr. Thornycroft has taken advantage of. The action is as follows: When the heat of the furnace gets to the lower part of the tubes the water above is driven forward, and a good part of it is sent over into the separator with the steam that has been formed; the steam being taken away to be used in the engine. The water which has been forced into the separator mixes with the cold feed water; the two supplies flow down the two downcomers into the wing cylinders, and thence rise again to be evaporated in the generating tubes, thus preventing the latter from becoming unduly heated, or any part of the steam from becoming superheated.

The circulation of water due to the cycle of events we have described is the prime factor in the success of the Thornycroft boiler. The rapidity of this circulation naturally depends on the difference between the specific gravity of the comparatively cold water in the downcomers, and the hotter mixture of water and steam in the generating tubes.

There are one or two details about the separator to which reference may be made. The baffle plate shown in Fig. 100—it is represented only by a line in the cross section Fig. 99—is for the purpose of directing the water brought over to the separator towards the bottom of that vessel, so that it

will not be sucked into the steam pipe. The serrated edges are of a form best calculated to let the water trickle off, thus destroying the momentum with which it was ejected from the pipe. The steam pipe itself is carried some distance inside the separator, and has a large number of saw cuts on its upper part to prevent an excessive rush of steam to any one point, a circumstance which would induce priming. The absence of straining through expansion and contraction will be evident from the formation of the generating tubes. Of course the tubes do expand when made hotter, that is a law of nature they must obey in common with all other boiler tubes. But then they have room to expand, and, supposing one wants to expand more than others, the only effect is that it becomes a little more bent.

A very exhaustive series of experiments have been made with the Thornycroft boiler by Professor Kennedy, from which the following details are taken. The boiler tested was that of a torpedo boat. The coal burnt was Nixon's navigation, having a calorific value of 14,900 thermal units per lb. This gives a theoretical evaporation of 15·41lb. of water from and at 212° Fahr. The boiler had a heating surface of 1837 square feet and a grate surface of 30 square feet; but in some of the trials the grate was partially bricked up. The engines were the usual type of triple compound generally used on torpedo boats, having cylinders 14, 20, and 31½in. in diameter by 16in. stroke; all cylinders were jacketed. There were separate engines for the circulating pump and the fan. There was also a separate donkey engine and steering engine. All exhaust steam was carried to the condenser.

The result of the trials are given in the table overleaf.

The results recorded show that boilers of this type are remarkably efficient as steam generators; indeed, we believe, as Professor Kennedy claims, that the trials form a "record" so far as the points dealt with are concerned. It would have been more satisfactory had the tests been of somewhat longer duration. That 13·4lb. of water can be fairly evaporated per lb. of coal (from and at 212°) is a revelation to many engineers, and the natural conclusion of some will be that a great deal of priming must have been going on. A careful examination of the figures does not tend to support this theory. In the first place the boiler was working at a remarkably easy rate when this duty was attained, burning only 7¾lb. of coal per square foot of grate per hour; whereas on another trial nearly 67lb. were burnt. If, in the first instance, there had been priming, in the second the engines would not have been able to have been run at all. It is to be regretted, however, that no practical means has yet been devised by engineers for determining the amount of priming on boiler trials.

	A. Natural draught.	D. Natural draught.	C.	B.	E.
Duration	5 hours 2 minutes	4 hours 57 minutes	5 hours 9 minutes	4 hours	2 hours
Boiler-pressure	186·00lb. per sq. in.	181·80lb. per sq. in.	171·20lb. per sq. in.	149·40lb. per sq. in.	180·50lb. per sq. in.
Air-pressure in stoke-hold	0·00	0·00	0·27in.	0·49in.	2·00in.
Total weight of coal put on fire	1680·0lb.	1006·5lb.	2877·0lb.	3575·0lb.	3503·0lb.
Total weight of ashes	270·0lb.	233·5lb.	197·0lb.	...	192·0lb.
Weight of ashes put back on fire	None	233·5lb.	170·0lb.	None	192·0lb.
Coal burnt per hour	334·0lb.	203·3lb.	559·0lb.	894·0lb.	1751·0lb.
Area of fire-grate	30 sq. ft.	26·2 sq. ft.	30 sq. ft.	30 sq. ft.	26·2 sq. ft.
Coal burnt per foot grate per hour	11·10lb.	7·74lb.	18·60lb.	29·80lb.	66·80lb.
Total feed-water used	...	11,291·7lb.	30,141·0lb.	34,332·0lb.	31,109·0lb.
Feed used per hour	...	2281lb.	5852lb.	8583lb.	15,554lb.
Feed temperature	78°·4	76°·3 Fahr.	78°·0 Fahr.	83°·8 Fahr.	111°·2 Fahr.
Steam	382° Fahr.	380°·2 Fahr.	375°·5 Fahr.	365°·5 Fahr.	379°·6 Fahr.
Water evaporated per lb. fuel under ordinary conditions, with ash utilised	...	11·22lb.	10·48lb.	[10·20lb.]	8·89lb.
Equivalent evaporation from and at 212° Fahr., with ash burnt	...	13·40lb.	12·48lb.	[12·00lb.]	10·29lb.
Equivalent evaporation from and at 212° Fahr. per lb. carbon, value in fuel	...	13·08lb.	12·18lb.	[11·70lb.]	10·04lb.
Temperature of gases in chimney	474° Fahr.	421° Fahr.	540° Fahr.	610° Fahr.	777° Fahr.
Air-pressure in chimney	0·00in.	0·00in.	+0·03in.	+0·12in.	+0·40in.
Total heating-surface	1837 sq. ft.	1837 sq. ft.	1837 sq. ft.	1837 sq. ft.	1837 sq. ft.
Ratio of heating-surface to grate	61·2	70·1	61·2	61·2	70·1
Water evaporated per sq. ft. of heating-surface per hour	...	1·24lb.	3·20lb.	4·70lb.	8·50lb.
Efficiency of boiler	...	86·8 per cent.	81·4 per cent.	78·2 per cent.	66·6 per cent.
Revolutions per minute	192·8	165·2	234·2	268·7	318·4
Steam-pressure in hp. valve chest above atmosphere	50·60lb. per sq. in.	22·70lb. per sq. in.	79·20lb. per sq. in.	120·70lb. per sq. in.	168·40lb. per sq. in.
Mean pressure, hp. cylinder	17·60lb. per sq. in.	13·00lb. per sq. in.	27·30lb. per sq. in.	37·60lb. per sq. in.	49·90lb. per sq. in.
Mean pressure, mp. cylinder	13·70lb. per sq. in.	9·50lb. per sq. in.	20·10lb. per sq. in.	27·20lb. per sq. in.	40·20lb. per sq. in.
Mean pressure, lp. cylinder	3·40lb. per sq. in.	2·20lb. per sq. in.	5·74lb. per sq. in.	8·29lb. per sq. in.	12·80lb. per sq. in.
Mean vacuum	28·04in. = 13·77lb.	27·97in. = 13·74lb.	28·34in. = 13·92lb.	28·06in. = 13·78lb.	25·25in. = 12·40lb.
I.H.P., hp. cylinder	41·7	26·5	78·4	124·1	195·3
I.H.P., mp. cylinder	66·8	39·6	118·7	185·1	323·6
I.H.P., lp. cylinder	41·8	23·0	85·0	140·0	255·8
I.H.P., total	150·3	89·1	282·1	449·2	774·7
Total feed-water per I.H.P. per hour	...	25·60lb.	20·74lb.	19·10lb.	20·08lb.
Jacket water per hour	151·0lb.	126·0lb.	236·5lb.	188·5lb.	292·5lb.
Jacket water per I.H.P. per hour	1·00lb.	1·43lb. = 5·60 per cent.	0·84lb. = 4·03 per cent.	0·42lb. = 2·20 per cent.	0·38lb. = 1·90 per cent.
Added feed-water per hour	31·7lb.	12·9lb. = 0·57 per cent.	44·6lb. = 0·76 per cent.	88·2lb. = 1·03 per cent.	145lb. = 0·93 per cent.
Fuel per I.H.P. per hour	2·220lb.	2·280lb.	1·981lb.	1·990lb.	2·260lb.
Carbon value per I.H.P. per hour	2·280lb.	2·334lb.	2·030lb.	2·040lb.	2·320lb.
Approximate speed of vessel on measured mile	11·8 knots	...	...	...	18·0 knots

Should any yachtsman be enthusiastic enough to test the evaporative efficiency of his boilers he must not be disappointed if the duty falls very far short of the 13·4lb. evaporated, as above quoted. In the first place he will not be evaporating steam from and at 212°; and it will be seen that under the conditions of actual work the evaporation was 11·22lb. of water per lb. of fuel per hour. The "from and at 212°" is a convenient standard used by engineers to bring different boilers working at different pressures to a fair level of comparison. The figure obtained is the result of calculation. In the next place the boiler was very much favoured, having an extremely light load put upon it. Even to get so excellent a result in evaporative efficiency as that quoted, it would not pay a yachtsman to carry about a big boiler which would be only evaporating 1¼lb. of water per hour for each square foot of heating surface, in place of the 4 to 4½lb., as in the case of the Meteor and Tartar hereafter quoted. It will be seen, too, by the table that although the boiler performed so well in this natural draught trial it was at the expense of the engine (perhaps it would be fairer to say the engines were too big for this low power), for the fuel economy was not so good in trial D. (natural draught) as in trials C. and B., when, however, the boiler efficiency was not so high. It should be stated here that these trials were not made to test the economy of the machinery as a whole, but solely to test the boiler.

If we have to look on trial D. as exceptional, trial E. may be taken as a fair test, although it might have been further prolonged with advantage. In this trial, when the draught was forced to 2in. water pressure, and nearly 67lb. of coal were burnt per square foot of grate per hour, the actual evaporation was nearly 9lb. per lb. of coal, equivalent to 10·29lb. from and at 212° Fahr. This may be taken as a better result than is usually obtained by ordinary marine boilers even when working with natural draught. The respective temperatures of the chimney gases for the two trials are worth noting. This evaporative result may be compared to the 7½lb. to 8lb. of water evaporated per pound of coal in the case of the four ships tested by the Research Committee of the Institution of Mechanical Engineers, as elsewhere quoted (see page 250), and is not far from the 9lb. to 10lb. of the best boilers tested at the celebrated Newcastle trials made by the Royal Agricultural Society in 1887. The fact that the Thorneycroft boiler was running at forced draught must not be forgotten, as this places it at considerable disadvantage in the comparison.

It may be well to point out in regard to trial D., that, although the boiler pressure was 181lb. the initial pressure in the high pressure cylinder was under 23lb., which would indicate that the steam was throttled.

This, of course, was against the engine (and also in some respects against the boiler), and this doubtless accounts for the low duty of the engine on this trial, as shown by the weight of feed water required for each unit of power—viz., 25·6lb. per I.H.P. developed; but it must be again repeated that these were not engine tests but boiler trials. There was also throttling in the other tests.

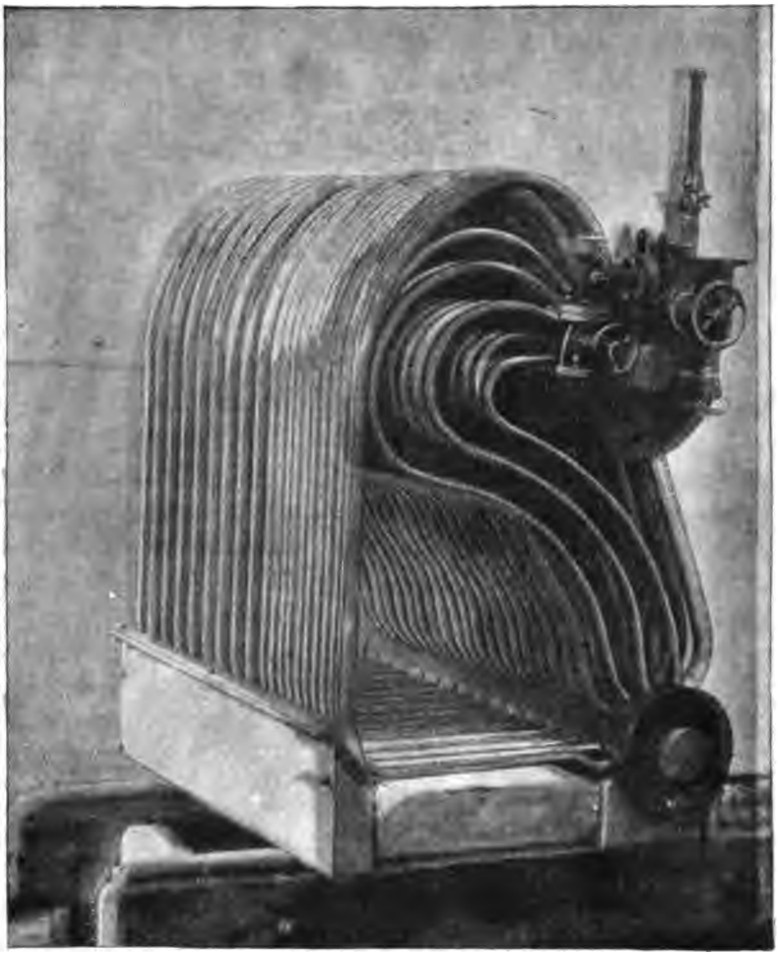


FIG. 102.

We have not data sufficient to enable us to state exactly how the Thornycroft boiler compares with the more ordinary types in the matter of weight and space occupied. In the former respect, however, it shows a marked superiority, and in the latter it has perhaps an advantage. The space occupied by the two Thornycroft boilers of the first class torpedo boat Ariete, referred to on page 267, was 33ft. of the length of the vessel,

or rather 33ft. was the length of the boiler compartment. The H.P. with forced draught being 1550 indicated. The engine-room was 18ft. long. With regard to weight, Mr. Thornycroft has stated at a meeting of the Institution of Civil Engineers, held 19th November, 1889, that the weight of this boiler, including water, was 12lb. per square foot of heating surface, as compared to 70lb. of the ordinary marine boiler. An official of the Danish Navy, who has had some experience with this boiler, gives the weight of three types of boiler, including water, mountings, and fittings, as follows: Low cylindrical 80lb., locomotive 33lb., and water tube (*i.e.*, Thornycroft) 29lb. per horse power developed at full power

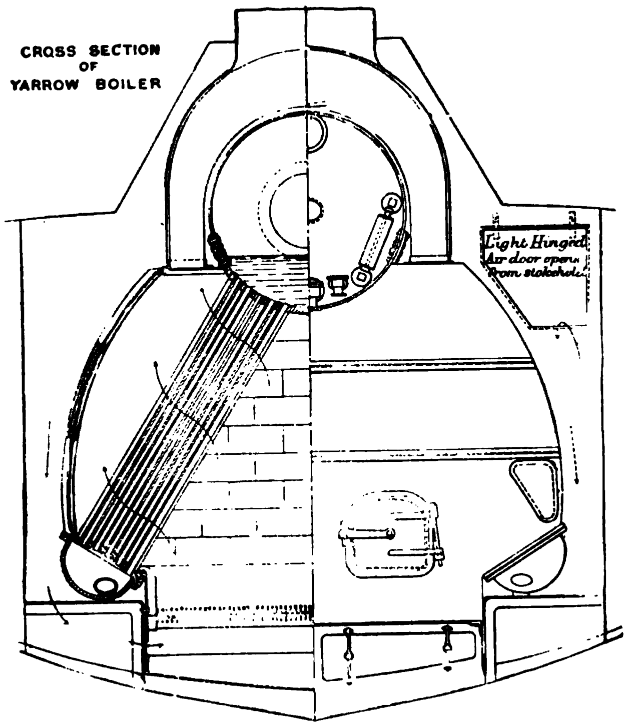


FIG. 103.

forced draught trials. Mr. Thornycroft gives the same figures in reference to a boiler weighing 9⅓ tons, which worked to 700 H.P. This, however, he stated was a higher duty than that boiler ought to have been put to.

Since the Speedy type of boiler—and it is to this type that reference has up to now here been made—was introduced, Mr. Thornycroft has modified the design in a way which enables a battery of boilers to be worked with the ship in a more compact manner than is possible with the orginal plan. This newer arrangement, which would be suitable for large yachts, is known as the Daring boiler, after the torpedo boat

destroyer, in which it was first tried. Another, and still later development, is the launch boiler, a simple, light, and compact design, in which the generating tubes are bent round and support the fire. In this way the problem of water fire bars, at which so many engineers have worked in times past, has been solved; for this type of boiler has worked satisfactorily during a fairly protracted test. An illustration of this boiler is given in Fig. 102.

The Yarrow boiler, as previously stated, differs essentially from the Thornycroft boiler. There is, however, some resemblance in the design, inasmuch as there is a top separator connected to two wing chambers by the generating tubes. On the preceding and following pages in Figs. 103 and 104 are given two views of the Yarrow boiler. The former is an end elevation partly in section, and the latter shows the water-containing part of the boiler, the casing, furnace, &c., being left out. Mr. Thornycroft, in designing his boiler, was especially careful to avoid straight generating tubes. He held that unequal expansion and contraction, due to local heating and cooling, would cause leakage at the tube ends, and here he was orthodox to accepted canons of scientific boiler design. Mr. Yarrow is, however, somewhat of an iconoclast, and in this, as well as in some other respects, he threw over orthodoxy. He further abandoned a second great principle in eliminating the down-comer pipes; the only communication between his top cylinder, or separator, and the lower wing vessels being by way of the generating tubes. Still another cherished convention was thrown overboard when he made the steam generating tubes deliver into the separator below water instead of above, as in the Thornycroft boiler. It will be seen, therefore, that the Yarrow and the Thornycroft designs represent two distinct types of water tube boiler; and as the differences between them embody some leading principles, it will be convenient to examine them at some length.

One of the first essentials to the successful working of a water tube boiler is efficient circulation. This has been already explained. The Yarrow boiler has no down-comer pipes provided especially for circulating purposes, but the back rows of steam generating tubes serve as downcorners. The action is as follows: When fires are first lighted the rows of pipes nearest the furnace get hot first, and the water in them, being thus made lighter, ascends, whilst the colder water in the back row of tubes descends. In this way an initial circulation of a comparatively feeble nature is set up. After a time steam is formed, first, of course, in the row of tubes nearest the fire. As circulation is due to difference in specific gravity of the contents of the upcast and downcomer pipes, it will be evident that as soon as steam is formed in the

generating pipes the circulation will be greatly accelerated, a mixture of water and steam being so much lighter than an equal volume of water only. Arguing on this basis the advocates of separate down-comers—and they are very numerous—maintain that the Yarrow boiler must be inferior to those types which have outside down-comers. They say that in the Yarrow boiler steam is apt to be formed in the back rows of tubes which act as down-comers. With high rates of combustion it is sure to be formed there. The difference in specific gravity of the two columns (*i.e.* the contents of the upcast and downcast pipes) is the cause

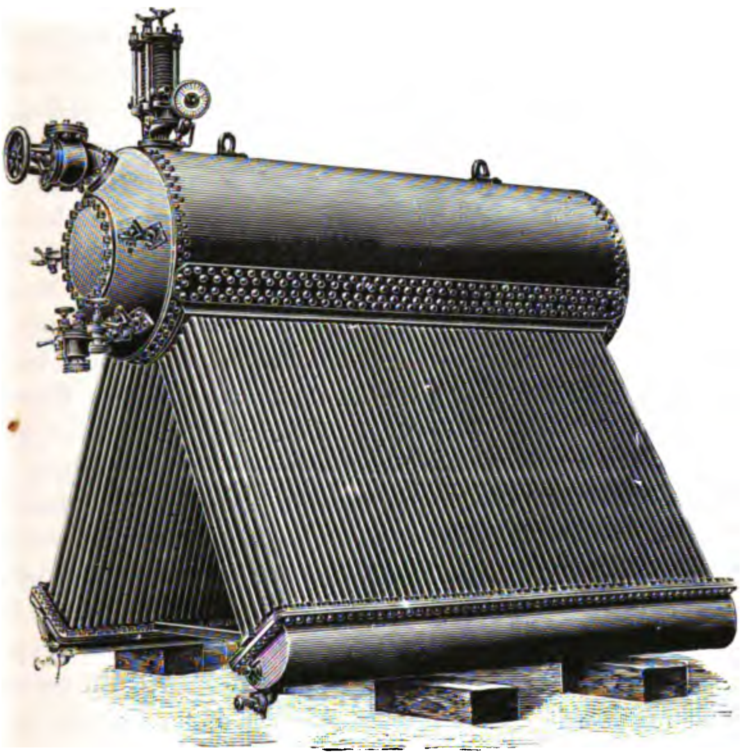


FIG. 104.

of circulation, and, as the presence of steam in the upcast pipe is the cause of this difference in specific gravity, any generation of steam in the down-cast pipes must detract from the vigour of the circulation, thus leading to a lower rate of evaporation and also to danger of the tubes becoming burnt.

The problem may be stated another way. The tendency of bubbles of steam contained in water is to rise, and in rising to carry water upwards with them. In the upcast pipes that leads to circulation, but when a bubble of steam tries to rise in a downcast pipe it struggles against the current and checks circulation. It is true a bubble in the downcast pipe

may be carried down by reason of the upcast pipe, on the whole, having more bubbles (circulation having thus been set up in a given direction), but, nevertheless, the said bubble does its best to check circulation, and succeeds to the extent of its power. All this seemed unanswerable for a time, until Mr. Yarrow exhibited some experiments which upset this and many other established theories. We have not space to even enumerate all the experiments shown, but one will serve our purpose. There was an apparatus consisting of a cylinder filled with water, from which depended two tubes communicating at their bottom ends, thus forming an inverted syphon. These tubes were partly of glass, so that the course of circulation could be followed. A number of Bunsen burners were applied to one tube, A, and none to the other, B. Steam was rapidly formed in the tube A, and circulation was thus set up, the current ascending in A and descending in B, just as it would in a boiler having an outside down-comer. An equal number of burners were then applied to tube B, thus converting the experimental apparatus into the semblance of a Yarrow boiler, but in place of the circulation being checked it was very greatly increased. The burners were then removed from the first tube, A, but those in connection with tube B were kept in place, and in spite of the fact that all the steam was being formed in tube B, that remained the downcast pipe, whilst tube A, in which no steam was being formed, remained the upcast pipe. Circulation was thus being carried on, apparently, in defiance of the law of gravity. Of course this was not so. The explanation is very simple, and goes far to illustrate the principles of circulation in water tube boilers. Taking the two tubes, A and B, which were of equal length, and remembering that circulation has already been established, we will imagine a bubble of steam to be generated in B (the downcomer) midway between the top and bottom ends. For half the length of the tube that bubble of steam is pulling against the circulation, that is to say it is trying to ascend, but the rush of water is too strong for it, and it is carried to the bottom of the inverted syphon. As soon as it gets round the corner into tube A it acts with the current in trying to ascend, and so enforces circulation. There is, however, this difference, that whilst it acted *against* the current when in tube B (where it was generated) for *half* the length of the tube, it acts *with* the current in tube A for the *whole* length of the tube,and thus there is a balance of accelerating power equal to half a tube available for maintaining circulation. The problem is a little complicated by certain minor conditions not here discussed owing to limits of space.

The facts above stated may be accepted as broadly applicable, and sufficient for practical purposes. To work the matter out fully in a scientific manner it would be necessary to take into consideration the

influence of varying pressure (due to head) on the steam generated at almost constant temperature, the effect of friction, the possible results of that very debatable phenomenon, surface tension, together with the viscosity of the fluid and many other conditions which would be necessary to a quantitative result but would not reverse the main conclusion. The task would be a most interesting one for any yachtsman having at his command abundant leisure, great perseverance, a well equipped physical laboratory, and the scientific attainments needful for a research so complicated.

The practical effect of the experiment quoted was to show that heating a down-comer accelerates circulation. To put the matter practically, it might be said, that as one must have pipes for the return current, why not utilize them as heating-surface if rather good than harm comes from the practice? These down-comer pipes cost money, occupy space, and mean weight. These are debit factors; let us get something from them for the credit side. To prevent misconception it should be pointed out that there are limits to this reasoning, though these limits are not reached in general practice. The maximum circulation would be attained when the difference in specific gravity between the two columns would be greatest, *i.e.*, supposing one column all steam and the other all water. With such a condition reached the subsequent formation of bubbles in the down-comer would be a retarding element, for these bubbles, while in the down-comer, would act against circulation, and could do nothing more to assist it after passing into the upcast pipe, as the latter would be already full of steam.

There are two other main points to notice in considering the relative positions of these two leading types of water tube boiler—namely, straight tubes as against bent tubes, and drowned tubes as against above water delivery. The former point may be dismissed very briefly. Undoubtedly the straight tubes do expand and bend when the boiler is steaming, and equally without doubt they do not always resume their absolutely straight form after cooling down. The true question is, however, whether this bending is injurious. In the first place it should be remembered that all boilers alter their shape when under steam. In shell boilers especially the stresses due to expansion and contraction are often very marked; in fact some of the greatest successes attained with torpedo boats in past times have been due to the fact that the loco-boilers have been so designed as to allow the parts to alter in shape at will—to "bellows" as it was called—rather than bind them up rigidly by stays, &c. Without doubt, flexibility is more apparent in the bent tube boiler than in the straight tube variety, but if the stresses put upon the straight tubes are not greater than the metal and the joints are calculated to bear easily, the objection of not making allowance for

expansion and contraction does not count for much. It would be useless to theorise upon this question. We have, however, a certain amount of practical experience, and so far as that goes, it would seem that the stresses put upon straight tube boilers are not more than the structure can bear. The writer can give one piece of personal testimony. The day after the trial of the Russian torpedo boat destroyer Sokol he made a close inspection of one of the Yarrow boilers fitted in that vessel. These boilers, it need hardly be said, had been tested in a way that rarely falls to the lot of yacht boilers, as the Sokol made on her trial what was then the highest speed on record. In the boiler referred to, however, although the tubes were very slightly bent out of the straight, there was not the sign of a weep at any of the joints; in fact, there was every indication that the structure was strong enough for the work required from it. Certainly the old loco-marine boiler would not have gone through the ordeal of the previous day with so little ill effect.

Bent tubes have, however, some advantages aside from the question of expansion and contraction. They give a larger percentage of heating surface for an equal number of joints. It is possible also to utilise a greater area of the barrels for tube attachment, as by bending the ends of the tubes they can always be brought in at any point radially. For this reason more rows of tubes can be inserted with the bent tube arrangement. On the other hand, the labour and consequent expense of bending is avoided with straight tubes, they are simpler, are more easily replaced, and, moreover, are open to examination and cleaning on the interior whilst in position.

It now remains to consider, the relative merits of drowned tubes, as in the Yarrow boiler, and above water discharge, as in the Thornycroft type. This subject takes us back incidentally to the problem of circulation. With drowned tubes the steam generated has to force its way through the water in the separator or top barrel. In other words, there is a head against circulation due to the height of water in the barrel. It should be remembered, however, that if there is a head exerting pressure hindering circulation above the upcast tubes, there is also a head assisting circulation above the downcast tubes, so that the two may perhaps neutralize each other so far as any pair of tubes may be concerned, but there are in ordinary working far more upcast than downcast tubes. With undrowned tubes the water in the top drum offers no resistance to the discharge of steam, but, on the other hand, all the unevaporated water carried round in the circulation has to be raised an additional few inches, so as to bring it in at the top of the drum. As, according to some experiments of Mr. Thornycroft's, the volume of water in circulation

is 105 times that of steam; this additional load on the energy available to produce circulation should not be disregarded.

The main objection raised against drowned tubes is that water may pass down any of them, and that thus an upcast tube may be converted into a down-comer, or the forces tending to produce circulation may be so closely balanced that it may be practically destroyed, and the tube become burnt. If this contention could be established it would be fatal to drowned tubes; but, so far as can be judged at present, the objection has little or no force. We must remember first that the difference in specific gravity of the contents of two or more tubes is the cause of circulation. We will suppose a tube to contain nothing but water. It would then be a downcast pipe, and would be naturally safe from burning. If, however, circulation be checked a large part of the water would be converted into steam, and, the specific gravity of the mass being thus lowered, an upward circulation would be set up. Mr. Yarrow's experiment, before quoted, illustrates how circulation reinforces itself, and this would lead to the supposition that, however feeble might be the initial flow in any one pipe, it would speedily become very rapid. Experiment has shown this to be the case. Among other tests made by Mr. Yarrow was one in which a full-sized section of one of his boilers was shown, the tubes having glass lengths let into them and the casing cut away so that observations could be taken. Steam was raised in this boiler, and the whole process of circulation was very prettily shown. At first there was the sluggish movement of the water due to the heating of the tubes nearest the fire, which, of course, carried the ascending current. After a time steam began to form, and then the circulation was much increased. As the hot gases penetrated to the rows of tubes further back, they in turn would become upcast tubes, until at last only the back rows acted as down-comers. The point to notice, however, was that in no case was the reversal of flow from downwards to upwards other than instantaneous; in fact from the time the boiler became thoroughly hot the circulation in every tube was rapid and vigorous.

The Thornycroft and the Yarrow boilers have been dealt with at some length, because they best illustrate fundamental principles, and it has been thought advisable here to deal with phenomena that are universal in application rather than describe individual examples of boiler design. At the present time new inventions are being brought out almost daily. A few of these possess merit, but it may be safely said that the majority are born only to die without leaving any mark. Of those express boilers which have achieved success practically the whole have been introduced in torpedo craft, and it may be said of the destroyers that

not one of them could have reached the remarkable speed they have attained had it not been for the water tube boilers.

Whether water tube boilers of the type already dealt with should be applied to yachts is a wide question upon which a reply of universal application cannot be given, as the conditions of working are so various. Each case must be considered on its merits. One of the first questions an owner should ask himself is: "How will his engineer regard the matter?" If the engine-room staff are against "new fangled notions" they "have not been brought up to," better stick to the old style or else get a fresh crew. The great question that remains to be solved is that of durability, and to attain this the owner must make up his mind to provide for an ample and sufficient supply of clean fresh water as "make up" for feed purposes. This can be done by distillers or by carrying fresh water in tanks. Water tube boilers are very nice in the matter of "feed." Unless this is attended to their constitutions rapidly deteriorate. Water tube boilers are easy to stoke, and the quickness with which steam can be raised without detriment to the structure is an especially valuable point in yachts. We have already made reference to the questions of space and weight occupied in regard to the Thornycroft boiler, and may add here some figures bearing on these subjects as affecting the Yarrow design. Messrs. Yarrow and Co. have recently supplied certain water tube boilers for some Dutch cruisers in which ordinary return tube boilers are also to be placed. There are in each ship eight Yarrow boilers and two return tube boilers. The two latter weigh, with water, 120 tons, the eight water tube boilers weighing 88 tons. In both cases firebars and fittings are included, but not chimneys. Without going into details it may be stated that the figures as to weight work out as follows: For each unit of power developed there will be required a little over 1cwt. (1·0666) of return tube boiler and water. With the Yarrow boilers the proportion of weight to power will be 0·234cwt. to one indicated horse power. At the time of writing the trials of these ships have not been made, but the results are calculated on well ascertained data. In regard to space the return tube boilers are each 13ft. in diameter by 10′ 6″ long, exclusive of uptake, which adds another 2ft. to the length. The Yarrow boilers are 9′ 3″ high, 9′ 3″ wide, and 9ft. long over casings. The top barrel extends, however, 1ft. farther at each end, the total length of the latter being 11ft. The Yarrow boilers are placed three abreast, occupy rather less fore and aft space and 3′ 9″ less height.

A water tube boiler for steam launches and torpedo boats has been introduced by the Liquid Fuel Company, East Cowes, Isle of Wight. These boilers have bent tubes somewhat of the Thornycroft form, and the furnace is so arranged that either petroleum, coal, or wood can be used.

All the pipes and other parts of the boiler are of copper, and with liquid fuel the boilers can be used for months without any cleaning of tubes, there being no soot deposit, as the combustion is practically perfect. The consumption of oil is from 1 pint to 1¼ pints per I.H.P. per hour. The cheapest kind of oil is generally used at a cost of about 5*d*. per gallon. The extreme lightness, efficiency, and easy management of this steam raiser has brought it into great request. The boilers are made of all sizes from 5 I.H.P. up to 500 I.H.P. If furnace bars are fitted, and coal used as fuel, there would be a reduction of about 15 per cent. in the I.H.P.

This liquid fuel boiler must not be confused with the oil motors, of which there are many in use, the best known being the Daimler (Messrs. Sumners and Payne, Southampton), the Vosper (Messrs. Vosper and Co., Portsmouth), and the Priestman (London). These motors have no steam generator, and are worked on much the same principle as a gas engine.

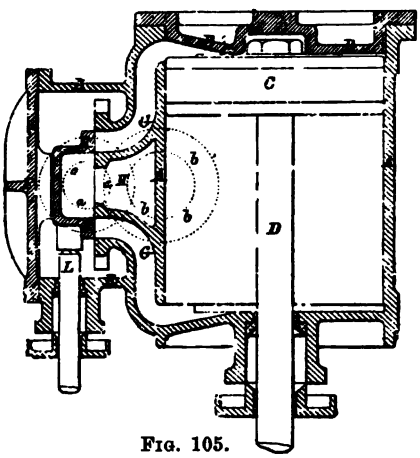


FIG. 105.

AA Cylinder.	EE Valve chest.	KK Slide valve.
BB Cylinder cover.	F Valve chest cover.	L Slide valve rod.
C Piston.	GG Steam ports.	*aaa* Position of steam pipe.
D Piston rod.	H Eduction or exhaust port.	*bbb* Position of exhaust pipe.

Of other types of boiler it is not necessary for us here to speak. The old vertical boiler has all but disappeared, although efforts are being made to introduce a new pattern of upright boiler, which is undoubtedly far superior to its prototype; as indeed it well need be.

Having given a short description of the steam generating apparatus, we will now proceed to deal briefly with the engine in which the steam is used. In the ordinary type of marine engine, the admission and eduction of steam to and from the cylinders is effected by means of a slide valve, which is caused to travel with a reciprocating motion on the cylinder face by suitable connections to the exterior working parts of the engine. By this action the two steam ports or passages are alternately opened and

closed, steam being thereby admitted into the cylinder above and below the piston in turn; by the same movement the eduction or escape of the spent steam is effected. Where a high rate of expansion is required, an additional valve, called an expansion valve, may be fitted; but, for the end we have in view a consideration of the ordinary single ported slide valve will be sufficient. It is not necessary for our immediate purpose to describe accurately and in all its details an engine as it would require to be arranged for actual work. There are many points which would only encumber our description of first principles, but which are absolutely essential to the design of a successful steam engine.

Fig. 105 represents a sectional elevation of the cylinder, slide valve, and valve case of an ordinary inverted yacht engine.

Fig. 106 is an outside view of the same cylinder, the valve chest cover

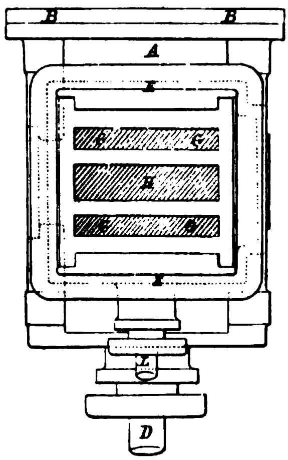


FIG. 106.

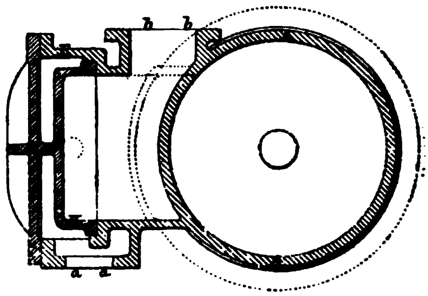


FIG. 107.

and slide valve having been removed, so as to show the steam and exhaust ports and the face upon which the slide valve travels.

Fig. 107 is a sectional plan of the same cylinder, slide valve, and valve chest. The same letters refer to the same parts in all the figures.

When it is required to start the engines steam is admitted from the boiler to the interior of the valve chest, and it must be remembered that the latter is constantly subject to the steam pressure as maintained in the boiler. The slide valve fits loosely on to the valve rod, and it is the pressure of steam at the back of the valve which keeps it close up to the cylinder face. Both the cylinder face and valve face are carefully planed in order to ensure the perfect contact of the two surfaces. In Fig. 105 the piston is at the top of the cylinder, and the engine would stand on what is called its top centre. The slide valve is covering both steam ports,

thereby cutting off all communication between the valve chest containing the steam at pressure and the interior of the cylinder. If the slide valve be now moved downward the upper steam port will become uncovered, and steam will flow from the valve chest into the upper part of the cylinder above the piston. The latter will accordingly be driven downwards, and assume the position illustrated in Fig. 108, which shows the position of the piston and slide valve, with the engine at half stroke, the upper steam port being open to the fullest extent, and the steam flowing freely into the cylinder. It is necessary, however, not only to admit steam to the cylinder, but to get rid of that steam after it has done its work. By reference to Fig. 108, it will be seen that the slide valve in descending has opened up a communication between the lower steam port and the exhaust port, so that the steam (which would have been admitted on the previous stroke to raise the piston, supposing the engine had already been at work) that had completed its task might find a means of exit. The passage of

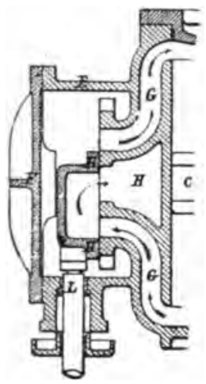


FIG. 108.

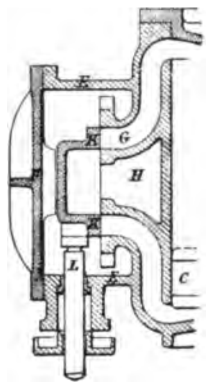


FIG. 109.

steam at pressure and of the exhaust steam is shown on the illustration by the arrows. The slide valve is now over both the lower steam port and the exhaust port, forming a cover to shut them off from the valve chest, but putting them into communication with each other, thus allowing the exhaust steam to pass from the cylinder through the steam port into the hollow of the valve, and from thence through the exhaust port to the exhaust pipe leading to the open air. The exhaust port is a passage cast in the metal of the cylinder, passing beneath the cylinder face, and communicating with the exhaust pipe. Referring again to Fig. 108, it will be seen that the piston C is in the centre of the cylinder. The engine crank is therefore at right angles to the line of travel of the piston,* in

* As a matter of fact, the crank would not be exactly at right angles to the piston in this position, owing to the obliquity of the connecting rod; but the description is sufficiently near the truth for our present purpose.

which position the engine would be said to be at half-stroke. The piston would, from that position, continue to descend until it reached the bottom of the cylinder. The slide valve has, however, completed its downward travel, and would at that time commence to ascend, and by the time the piston is at the bottom of the stroke, as in Fig. 99, the slide valve will have completed half its upward motion, and will be covering the two steam ports, shutting off the cylinder from communication with both the steam chest and exhaust port. In a single cylinder engine the momentum of the fly-wheel would now carry on the action for the short time that there would be no steam in the cylinder, and the slide valve, continuing to ascend, would uncover the lower steam port. Through the latter steam would flow, and the same action would follow on the upward stroke as has already been explained, excepting that the motions would be reversed, and the upper steam port would form the passage for the exhaust steam. That the piston and slide valve do not travel upwards and downwards simultaneously is obvious. In an engine arranged as the one we are now considering (which, as has before been remarked, is not one exactly suited for satisfactory working), whilst the piston is making its downward stroke the slide valve is performing the second half of its descending stroke and the first half of ascending stroke; the remaining half of the ascending stroke and its first half of the descending stroke being made while the piston is moving upwards.

The means by which the reciprocating motion of the piston is converted into the rotary motion of the shaft, through the cross head, connecting rod, and crank, will be sufficiently plain to anyone examining an engine to obviate the necessity of an explanation. The action by which the rotary motion of the shaft is again converted into a reciprocatory movement in the slide valve is not so obvious, and a consideration of this point will bring us to the questions of "lap" and "lead," two very important features in steam-engine economy. In by far the greater number of marine engines the slide valve is actuated by means of one or more eccentrics. For the present we shall not discuss the question of double eccentric and link motion, by which most engines are reversed, but will suppose our engine to be fitted with a single fixed eccentric.

By referring to Fig. 105 it will be seen that the slide valve when at the middle of its stroke exactly covers the two steam ports, and that any movement either upwards or downwards will open one of the ports and admit steam, putting the other port in connection with the exhaust port. Upon consideration of the movements of piston and slide valve, as illustrated in Figs. 105, 108, and 109, it will be seen that steam is only shut off from entering the cylinder as the piston reaches the end of the stroke or, as

an engineer would say, "steam follows full stroke;" and that steam is allowed to escape from the other end, also until the end of the stroke. This is an arrangement very undesirable in actual practice; first, because the whole fabric of the engine would be violently jarred on each reversal of the motion of the piston; and, secondly, the steam would be used very wastefully, as no advantage would be taken of its expansive property. As a matter of practice, in simple marine engines not fitted with expansion valves steam is usually admitted only whilst the piston is making from half to seven-tenths of the stroke, the rest of the work being done by the expansion of the steam in the cylinder.

Fig. 110 shows the same cylinder illustrated in Figs. 105 to 109, but the valve has been altered by an additional piece being added to it at the top and bottom. In Fig. 110 these pieces are shown by the unshaded portions. The strips added extended the whole width of the valve, and, for the sake of illustration, we will suppose they measure one inch each

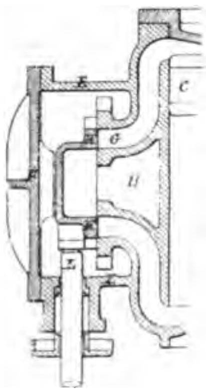


Fig. 110.

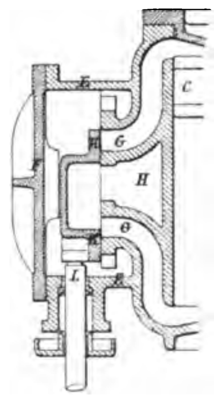


Fig. 111.

vertically.* These added pieces are the lap of the valve, being the extent that the valve overlaps the ports when in its central position.

In Fig. 110 the piston is on the point of completing the upward stroke, and, were there no lap on the valve, steam would have been admitted up to the end of the piston stroke. The addition of lap, however, has caused the port to close some time sooner, and the last part of the upward stroke must have been performed without an influx of steam to the cylinder, having in fact been made by the elasticity or expansive property of the steam. The work obtained from expansion would have been lost, through the steam escaping by the exhaust port at almost the same pressure as it entered the cylinder, had steam been carried full stroke.

* The illustrations here used are reduced to scale from the working drawings of an 18in. cylinder marine engine with 16in. stroke. The lap of the valve was 1in., and the stroke of the valve 3¾in. The throw of the eccentric would therefore be 1⅞in.

There is yet another alteration to be made in the arrangement in order that the value may be properly adjusted. Referring again to Fig. 110, it will be noticed that the piston has completed the upward stroke and requires steam above it to make the descending motion. The valve, however, has yet to travel downward 1in. (the amount of the lap) before the upper port will be opened. Unless, therefore, there were a heavy fly-wheel on the engine, which, by its momentum would carry the working parts forward, the machinery would come to a stop. To obviate this it will be necessary to shift the position of the eccentric on the shaft. In Figs. 108, 109, and 110 the crank and eccentric are set at right angles to each other, and whilst there was no lap to the valve, this arrangement answered well enough. Under the different conditions of working introduced by the addition of lap to the valve, it will be necessary to alter the relative positions of crank and eccentric from right angles to a larger angle. In Fig. 111 the eccentric has been shifted forward, and firmly keyed to the shaft in its new position, the effect of this having been to bring the valve down one inch, so that it will be on the point of opening the exhaust port when the piston is at the top of the stroke. The same movement will naturally result on the commencement of the up stroke when the piston is at the bottom of the cylinder. The eccentric must, however, be set forward a little more than the distance due to lap, in order to form the "lead" necessary to the quiet and regular running of the engines. By this means steam will be admitted above or below the piston just before the end of the stroke. This steam forms an elastic cushion or buffer, which brings the piston gently to rest for the instant of time that is required for the reversal of the reciprocating movement. This is the process that is referred to by the term "cushioning of steam." Cushioning may also be caused by early closing of the exhaust; but this point need not be considered now.

Having considered the action of the steam in the cylinders, we will now proceed to follow it up after it has passed the exhaust port. In non-condensing engines the exhaust pipe generally discharges into the chimney, in order to force the draught. Nearly all the marine engines in the present day, however, are surface-condensing. The weight of the ātmosphere at sea level equals roughly about 15lb. to the square inch, and the exhaust steam escaping from the cylinder of a non-condensing steam engine has to displace an equal volume of atmospheric air, thus giving a constant back pressure on the piston equal to the atmospheric pressure. In cases where the nozzle of the exhaust is reduced for forcing the draught, the back pressure will be proportionately greater. If in place of allowing the exhaust steam to escape into the air we conduct it to a closed vessel, and

surround this vessel with a constantly running stream of cold water, the temperature of the steam will be lowered to such a degree that it will become liquefied; and as water only occupies a sixteen hundred and forty-second part of the space which would be filled by an equal weight of steam at the atmospheric pressure, the remaining space in the condenser becomes void, so that the exhaust steam, in place of having to force its way into the atmosphere, finds an unretarded exit into the partial vacuum of the condenser. It will not, of course, be understood that with condensing engines steam is usually discharged from the cylinder at atmospheric pressure; as a matter of fact it is considerably less. In consequence of air leakage through the joints and connections of even the best-made machinery, the vacuum may be affected; in addition to this, a certain quantity of air is present in all water in a natural state, and this air, coming over with the steam, expands and further impairs the action. The value of the vacuum obtained in this manner is usually expressed in inches on the barometer; 2in. equalling roughly a pressure of about 1lb. per square inch.

Marine condensers are of two kinds, viz., of the jet or of the surface type. The condenser is composed of a hollow vessel, into which the exhaust steam passes, and is there condensed by a spray or jet of cold water being injected into it. For boats running in fresh water the plan does not possess the disadvantages that would be present in vessels used at sea, when the condensing water would necessarily be salt, and, mixing with the exhaust steam, would be pumped into the boiler as feed.

Surface condensers are made in many forms, but they all act on the principle of abstracting the heat from the exhaust steam by means of a stream of cold water passing over the surface of metal vessels or tubes, the steam being on one side of the metal and the condensing water on the other. By this plan the condensing water and the water condensed from the steam are not brought into contact, so that the fresh water is obtained for feeding the boiler even if the refrigerating water be salt.

The ordinary marine surface condenser is composed of a hollow vessel, which sometimes takes the form of a drum or cylinder, closed at the ends, and which has a number of tubes running through it in a line parallel to its axis, and projecting through the tube plates. Water is drawn from the sea by a suitable pump, is forced through the tubes, and afterwards escapes overboard. The exhaust steam from the engine enters the interior of the drum, and is condensed by being brought into contact with the cold surface of the tubes. This arrangement is frequently reversed, inasmuch as the steam may be passed through the tubes whilst the condensing water circulates between them; but it is obvious that the principle

of action will be the same in either case. In the best-arranged machinery the circulation of water in the condenser is effected by means of separate engines working a centrifugal pump. The advantage of this arrangement is that the pump may be kept at work, and the condenser so cooled, whilst the main engines are at rest, and a vacuum is thereby formed immediately the exhaust steam from the main engines is admitted to the condenser upon the vessel getting under way. It is customary in the larger class of engines to arrange the condenser so that it will form a part of the standard of the main engines. In some high-speed vessels, such as torpedo boats, the circulation of the condensing water is caused by means of the speed with which the vessel travels: a short scoop projects through the skin of the boat below water, the orifice pointing forward; as the vessel moves ahead at a rapid speed, the water rushes in, and by its momentum is carried through the condenser. A steam jet is supplied, acting on the principle of the injector, to promote circulation when the vessel is at rest. Messrs. Yarrow and Co. have a compound condenser in which the refrigerating water is supplied partly by the passage of the boat through the water, and partly by a separate pump.

A surface condenser, formed by a length of tube, placed outside the vessel below water, is much used, even for vessels of considerable size, in the United States and Australia. The following remarks are taken from an official report, made by Mr. Isherwood, of the United States navy, on two vessels fitted in this way:

This arrangement makes the lightest and most efficient surface condenser possible, and is vastly superior to an inboard surface condenser, avoiding the numerous joints between the tubes of the latter, the weight and bulk of its shell, the weight of the tube plates, and weight and bulk of a pump to supply refrigerating water, the weight and bulk of the injection and outboard delivery valves, and the possibility of air-leaks. The outboard condensing pipe has a maximum supply of refrigerating water of the temperature of the seawater on every portion of the pipe thereby requiring much less condensing surface than in the case of an inboard surface condenser, over whose tubes the same refrigerating water passes successively from one end to the other with continuously increasing temperature, and consequent decreasing condensing efficiency. In large vessels which would require docking for an examination or repair of the outboard condensing pipe, such an arrangement would not be judicious; but to a small vessel like a launch, cutter, or yacht, easily taken out of the water anywhere, this remark does not apply. The resistance of the vessel is, of course, increased by the outboard condensing pipe, due to its projecting cross section and its surface, but the power required to overcome these additional resistances is much less than what would be needed to work the pump supplying the refrigerating water to an inboard surface condenser.

To this we would add that the outboard condenser should be far less costly than the ordinary condenser with its circulating pump, and it can be placed so as to be protected by the keel in the event of the vessel grounding on a fair bottom.

The water of condensation is drawn from the condenser, together with such air and uncondensed vapour as may be present, by means of the air

pump, and is delivered into an open vessel called the hot well, from whence it is taken by means of the feed pump to the boiler, where it will be re-evaporated.

Returning to the action of the steam in the cylinder, we will first describe a very simple method of finding the point at which steam is cut off, supposing the lap, lead, and length of travel of the slide valve are known.

Describe the circle A B C, Fig. 112, the diameter of which equals the travel of the slide valve, and with the centre of this circle as a centre describe the circle D E, the radius of which equals the lap of the valve. With the point B as a centre describe a circle, the radius of which, B G,

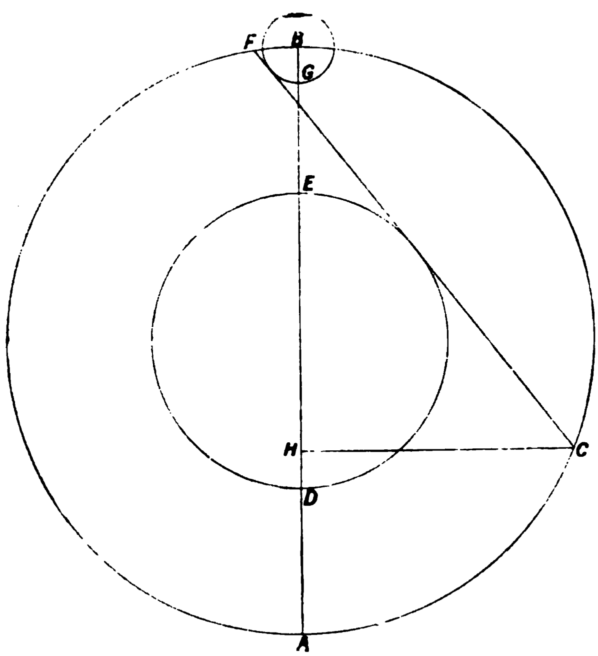


Fig. 112.

equals the lead of the valve. Draw the straight line F C a tangent to the circles G and D E. From the point C, where the line F C cuts the circle A B C, draw the line C H at right angles to A B. The point H will be the mean point cut off of the valve, A B representing the whole travel of the valve. This graphic method of determining the point of cut off of a common slide valve obviates a great deal of calculating, and, as for general purposes the mean cut off is what is required, the diagram is simplified by neglecting the obliquity of the connecting and eccentric rods.

Leaving out the disturbing influence of "port and clearance," the ratio of expansion of steam is determined by the space in the cylinder swept

by the piston up to the point of cut off, compared to the total space swept by the piston. Whether the expansion takes place in one cylinder, as in simple engines, or in two or more cylinders on the compound system, the principle is the same. Thus, if we have a cylinder of any given capacity with a slide valve arranged to cut off steam at half stroke, the ratio of expansion would be 2, for the space occupied by the steam at the end of the stroke would be twice that which it occupied at the point of cut off. If, however, for the purpose of further expanding the steam, the engine should be compounded by adding a second cylinder, say of three times the area of the first, but of the same length of stroke, the ratio of expansion would be 6; for the steam would ultimately become expanded into six times its original volume at the point of cut off in the high-pressure cylinder, having doubled its volume in the high-pressure cylinder, and again trebled this volume in the low-pressure cylinder. In this case no allowance has been made for steam in passages and clearance spaces, or for condensation in the cylinder, details, however, which materially affect actual work done.

The pressure of steam is approximately in the inverse ratio to its volume. For example, a cubic inch of steam at 40lb. pressure will be reduced to 20lb. pressure if expanded into two cubic inches, or 10lb. pressure if into four cubic inches. By the foregoing explanation, it will be seen that, given the cross sectional area of cylinder, length of stroke of the piston, initial pressure of steam in the cylinder, and the amount of lap and lead on the slide valve, we can determine theoretically the amount of steam consumed per stroke, the ratio of expansion, and the terminal pressure of steam in the cylinder. The calculations will not of course give practically correct results; for the absolute work done by the steam, it will be necessary to take indicator diagrams from both cylinders, and the nearer the results so attained approach the theoretical results, the better will be the design and workmanship of the machinery.

For the purpose of giving a practical illustration of what has been stated, we will calculate theoretically the ratio of expansion of steam in the engines of a small yacht we recently had under trial. The initial pressure, or pressure at which steam entered the high-pressure cylinder, was 62lb. per square inch above the atmosphere, or 77lb. absolute—that is, above zero. Steam was admitted for seven-tenths of the stroke when it was cut off by the slide valve, and the rest of the work performed by it, both in the high and low pressure cylinders, was due to expansion. The diameter of the high-pressure cylinder was 17in., and the stroke or space swept by the piston 21in. Taking the area of a circle of 17in. in diameter, and multiplying it by 21, we get the content of the cylinder in cubic inches.

The area of the cylinder in this case is 226·9 × 21 = 4764·9, which is the capacity of the cylinder in cubic inches. Steam, however, was only admitted for seven-tenths of the stroke = 14·7in. If we multiply this by the area of the cylinder in square inches, the product will be the cubic inches of steam at the initial pressure taken by the engine at each stroke, 14·7 × 226·9 = 3335·43. By dividing this by the capacity of the cylinder in cubic inches we get the ratio of expansion in the high-pressure cylinder $\frac{4764 \cdot 9}{3335 \cdot 43} = 1 \cdot 428$, which was the ratio of expansion in the high-pressure cylinder alone in the engine under consideration.*

To ascertain the total expansion of steam in both cylinders we find the proportion the capacity of the low-pressure cylinder bears to that of high-pressure cylinder, and multiply it by the ratio of expansion in the high-pressure cylinder. In the present case the capacity of the low-pressure cylinder is 16,888·2 cubic inches, which, divided by 4764·9 (cubic inches in high-pressure cylinder) = 3·50, being the proportion of low to high-pressure cylinder. By multiplying this by the ratio of expansion in the high-pressure cylinder we get the total expansion, 3·50 × 1·42 = 4·97, which is the number of times the steam is expanded in both cylinders, or the total ratio of expansion. If we divide the initial pressure by the total ratio of expansion, we get theoretically the terminal pressure of the low-pressure cylinder.

So far we have treated of a few elementary principles of the steam engine. The illustrations have been divested of all details not absolutely necessary for explanation, and therefore accurate results would not be obtained by estimating the power of engines simply on the rules here laid down. It is by the study of diagrams taken by means of the indicator that a satisfactory knowledge of the work done by the engine, and the arrangement of the machinery, whether good or bad, can be obtained.

The steam-engine indicator, the original introduction of which is due to the prolific genius of James Watt, has done more towards advancing the science of steam engineering than any other invention; indeed, it would hardly seem unreasonable to question whether steam engineering could be classed as a science at all but for the aid of this little instrument, without which much that is now plain to us would have been unknown. The primary use of the indicator is to measure the power exerted by the steam-engine operated upon. The power thus measured is known as the indicated horse-power, and is the result of the total power developed by the steam from the time it passes into the engine until it is finally delivered to the

* This is a low rate of expansion, and the engines would have been improved by an earlier cut off.

exhaust port. The total indicated horse-power of an engine is by no means the power that is available for the propulsion of a vessel, as the friction of the engine and the driving of the different pumps, &c., absorb a certain amount of energy. To find the actual power available for propelling the vessel, recourse must be had to the use of the dynamometer, the application of which is both costly and troublesome, whereas indicator diagrams can be taken without disturbing the ordinary running of the engines. These diagrams also give indications of any defects arising from faulty side-valve arrangement and consequent bad distribution of steam; what proportion the initial pressure of steam in the cylinder bears to the boiler pressure, and consequently whether there is any "wire-drawing" on account of insufficient area of steam pipe or cylinder ports, &c.; whether there is leakage of steam past the piston, or undue liquefaction in the cylinder; the quality of vacuum; in fact, to an experienced engineer, there is scarcely a fault that can occur in the motive part of a steam engine to which a clue will not be revealed by a study of the indicator diagrams. Having said this much, it can hardly be necessary to impress on the amateur engineer the utility of mastering the use of the indicator, and the following notes are penned with a view to assisting to this end.

The Richards indicator being still the standard instrument (although some excellent indicators have been introduced of late for special high speed work), it is to this that reference will be made in the following notes, and which is illustrated in Fig. 113. The indicator is attached at A to the cylinder of the engine to be operated upon, or more generally to the branch pipes leading from the cylinder. B is a cock, by opening which a communication is established between the engine cylinder and indicator cylinder CC. In the illustration, one side of the cylinder is removed in order to show the internal arrangement. D is a piston fitting the cylinder sufficiently close to prevent serious leakage of steam, but at the same time working easily enough to obviate any appreciable friction. Above the piston is placed a coiled spring, so arranged as to exert a certain definite resistance in the event of the piston rising or falling. A piston rod works through the cylinder cover, being attached, by means of the link E at its upper extremity, to the lever F, the fulcrum of which is, at G, carried on the bracket H. The end of F is attached to the link I, and the latter is again connected to the radius arm J working on the bracket K. In the centre of the link I, and at right angles to it, is placed a pencil, L, composed of a small piece of brass wire. M is a drum, upon which the paper is wound which is to receive the diagram. When the apparatus is working, this drum makes a partial revolution on its axis, but when there is no strain

on the cord N, it is kept in a certain definite position by means of a spring in the interior and stop on the outside. N is a cord, one end of which is fastened to the drum, the other end being carried away to attach to some working part of the engine. The cord makes one turn round the drum,

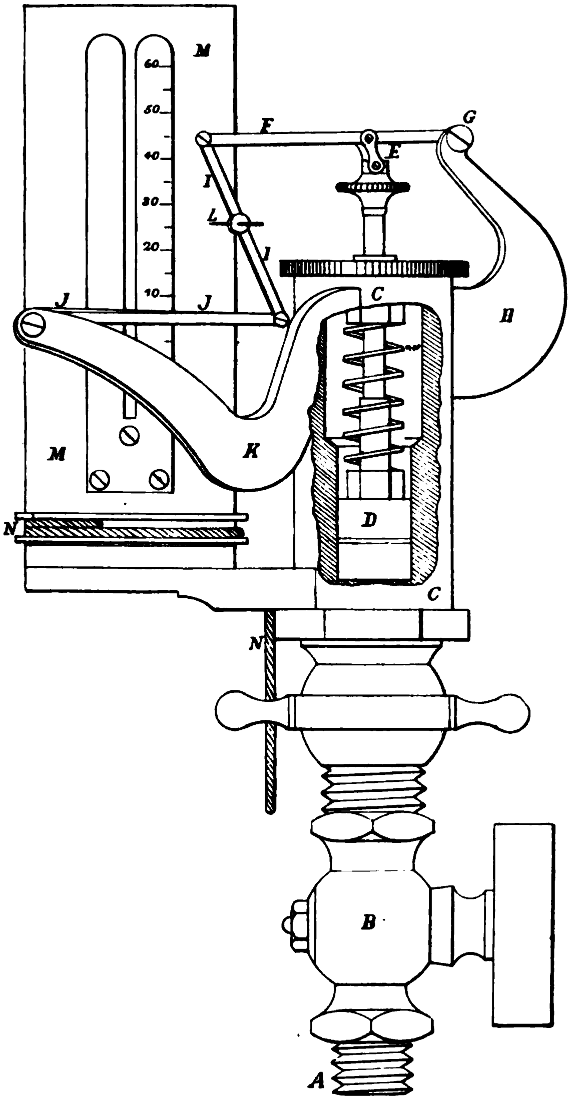


FIG. 113.

as may be seen by reference to the illustration. The diagrams are taken on paper specially prepared for the purpose. This paper is mounted on the drum by slipping it under the narrow strip of metal on which the scale is marked, as shown in the figure. The area of the indicator piston in the Richards Indicator is half a square inch; and it will be

obvious, that, when the cock B is open, whatever pressure there may be in the engine cylinder and pressing on the engine piston, there will also be a corresponding pressure on the indicator cylinder and pressing upwards the indicator piston. Thus, if the pressure in the engine were 80lb. to the square inch (or 40lb. to half a square inch), the indicator piston would receive an upward pressure equivalent to 40lb. The spring above the piston is made of such a strength that it will be compressed, and allow the piston to travel upwards (or be extended downwards if subject to vacuum), a certain defined distance for every pound pressure that is put upon it. Each indicator is provided with a number of springs, suitable for the varying pressures of steam that may be used. These springs are interchangeable.* Whilst the indicator is in use the piston may be said to be subject to two forces, that resulting from the pressure of the spring above, and that resulting from the pressure of the steam below. In the event of there being a vacuum in the engine cylinder, the spring will be in tension, and the indicator piston will therefore be drawn downwards. For the sake of simplifying our explanation, we will at present refer to non-condensing engines only, in which the pressure in the cylinders would not fall below the atmospheric pressure. We shall consider the subject of vacuum later on. Supposing, now, it were requisite to take indicator diagrams from an engine, the maximum pressure in the cylinder being about 80lb. to the square inch. We should place in the indicator spring No. 7, which is adapted to register pressures from that of the atmosphere up to 100lb. per square inch, and will be compressed one-fourth of the thirty-second part of an inch for each ½lb. pressure that the piston is subject to; it being remembered that the indicator piston is half a square inch in area, and therefore the total pressure on the piston is only equal to half the registered steam pressure. The motion upwards of the indicator piston will therefore be the one hundred and twenty-eighth part of an inch for each pound pressure on the square inch in the engine. The piston, in travelling upwards, will carry with it the lever F, and, by means of the link I, the pencil L. The latter, however, will rise four times the distance traversed by the piston, as F is a

* The springs are made to ten scales, as follows:

No 1.	$\frac{1}{8}$ inch on the scale represents 1lb. pressure on the square inch, indicates from	—15 to	+	10
„ 2.	$\frac{1}{12}$	—15 „	+	22
„ 3.	$\frac{1}{16}$	—15 „	+	35
„ 4.	$\frac{1}{20}$	—15 „	+	47
„ 5.	$\frac{1}{24}$	—15 „	+	60
„ 6.	$\frac{1}{30}$	—15 „	+	80
„ 7.	$\frac{1}{32}$	Atmosphere „	+	100
„ 8.	$\frac{1}{40}$	„	+	125
„ 9.	$\frac{1}{48}$	„	+	150
„ 10.	$\frac{1}{56}$	„	+	175

lever of the third order, the power being at E, and the arms of the lever are in the proportion of four to one. The reason this greater travel of the pencil is given is in order that a fair-sized card may be taken, with comparatively small movement of the spring and working parts of the indicator. In the original instrument the pencil was attached rigidly to the piston rod by means of a short arm, the paper drum being placed higher and closer to the cylinder for the purpose. This plan answered well enough so long as low pressures and a small number of revolutions per minute were required to be recorded; but the higher pressures and quicker-running engines of more recent times rendered the diagrams taken with the original type of indicator useless. This followed from the vibration caused in putting in motion the long spring required in order to get a diagram sufficiently large when the pencil had only the same travel as the piston; the momentum of the heavier moving parts necessary in the old instrument at the same time adding to the defect. The short stiff spring and lighter reciprocating parts of the Richards indicator will allow of results being obtained from engines working up to the highest pressures now in use, when running at very considerable piston speeds. In some of the extraordinary high-speed engines, however, that have been introduced within the last few years, such as those in torpedo boats, even the Richards Indicator has not been found sufficient, and a new instrument, the invention of Mr. Darke, and also another, the Crosby Indicator, which is on the whole best adapted to very high speeds, have been introduced. To return to our explanation. The arm J, the lever F, and the link I form a parallel motion for the pencil L; that is to say, by this arrangement the pencil is moved in a parallel with the line of travel of the piston. The arms K and H are mounted on a sleeve or outer covering to the cylinder, which will slide or partially revolve on the latter, so that the whole parallel motion and the pencil attached can, by hand, be withdrawn from or pressed against the paper at will; or the cylinder is made in two parts with the same object. So far we have described the manner in which a vertical line drawn on the card by the pencil will demonstrate the highest pressure of steam in the engine. It is, however, the office of the indicator to record the steam pressure within the cylinder at every portion of the stroke. In order to accomplish this, the cord N is fastened to some working part of the engine, which makes its stroke simultaneously with that of the piston. In simple non-condensing engines the main cross-head is the usual point of attachment. It will be remembered that the cord makes one turn round the drum on which the paper is mounted; it will therefore be seen that, when the string is drawn downwards by the working of the cross-head, the drum will make one revolution or partial revolution on its axis, and on

the return stroke of the engine the string will be slackened, and the drum will be brought back to its original position by the spring already referred to as being contained within it. When the string is attached to the engine cross-head, the motion has to be reduced to suit the circumference of the paper drum, otherwise the string would be broken.

It has thus far been shown that there are two motions made by the indicator—the first being that of the pencil upwards or downwards in a vertical direction, corresponding to pressure; and the second that of the paper on the drum, representing the stroke of the engine.

In order to render our illustration as simple as possible, we will suppose an engine arranged as in Figs. 105 to 109, pp. 211 and 213. This

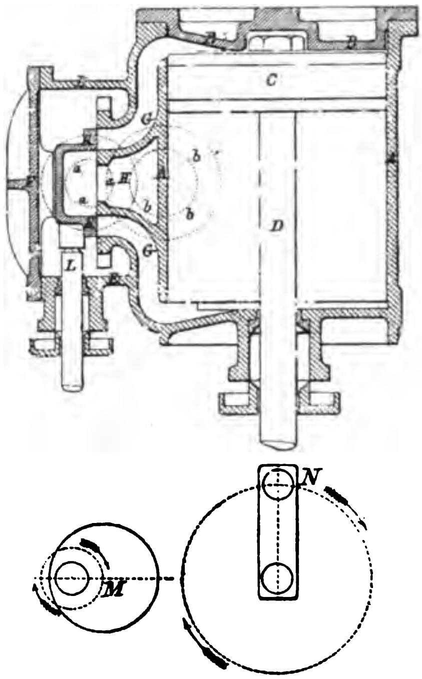


Fig. 114.

AA Cylinder.
BB Cylinder cover.
C Piston.
D Piston rod.
EE Valve chest.
F Valve chest cover.
GG Steam ports.
H Eduction or exhaust port.
KK Slide valve.
L Slide valve rod.
aaa Position of steam pipe.
bbb Position of exhaust pipe.
M Position of slide eccentric.
N Position of crank.

engine, it will be remembered, had neither lap nor lead to the valve, steam being admitted at the beginning, and carried to the end of the stroke, the eccentrics being set at right angles to the main crank, and the slide valve exactly the same length as the distance between the outer edges of the steam ports (see Fig. 114).

The diagram obtained from such an engine (putting aside all the

subsidiary and accidental variations which are inseparable from the working of any engine) would be a parallelogram such as Fig. 115.

The diagram is supposed to be taken from the top end of the cylinder. When the indicator pencil was at the point *a* on the card, the engine piston would be at the top end of the stroke. The slide valve would have completed half its downward stroke, and would, therefore, be just on the point of opening the upper steam port, as shown in Fig. 114. Immediately upon the slide valve opening the upper port G, steam would pass into the engine cylinder, and from thence to the indicator cylinder, causing the indicator piston to fly upwards, and the pencil to trace the line *a b*. This is termed the admission line of the diagram. So long as the steam port G remained uncovered, the pressure of steam would be maintained in the

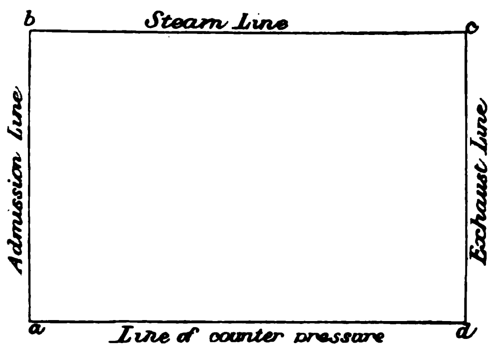


FIG. 115.

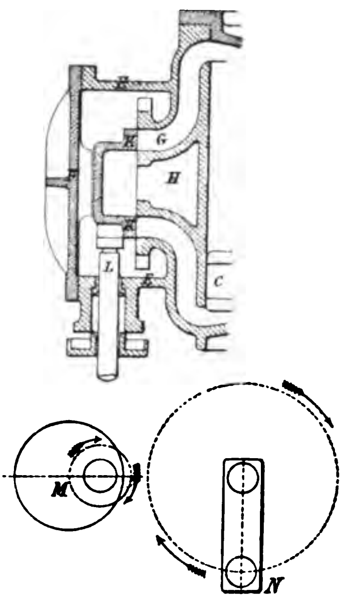


FIG. 116.

engine, and the indicator piston would be pressed upwards against the reaction of the spring, and the pencil would continue in the same position. The drum of the indicator, on which the card is mounted, would, however, revolve, owing to the movement of the string attached to the engine crosshead, and this would result in the line *b c* on the diagram, which is known as the steam line. This in the present case exactly represents the downward travel of the engine piston. By the time the pencil had reached the point *c*, the piston would be in the position shown in Fig. 116, having completed its downward stroke.

The slide valve is now ascending, and at this point will simultaneously shut off the flow of steam through the upper port and put the latter in

communication with the exhaust port H. The pressure above the engine piston will be released, and it will thence follow that the spring above the indicator piston will instantly force the latter to its original position (the steam pressure which upheld it being removed), thereby causing the pencil to trace the line *c d*, called the exhaust line. At the point *d* the cross-head to which the string is attached will be at the bottom end of the stroke, and from thence it will ascend, thereby slackening the cord, and allowing the spring within the paper drum to reverse the motion of the latter. This will produce the line *d a*, called the line of counter-pressure, or more usually back-pressure.

It will be seen by the foregoing explanation that all upward vertical lines are due to rising pressure; descending vertical lines are traced by falling pressure; and horizontal lines denote unvarying pressure. It must also be remembered that vertical lines are the result of movement of the pencil and horizontal result from movement of the paper on the drum. Curved or inclined lines would indicate a simultaneous movement of pencil and drum, *i.e.*, the pressure would be varying during the progress of the stroke of the engine. There is nothing in a diagram in itself to indicate whether it has been taken from the top or bottom of the cylinder; but it is usual for the admission line to be on the left of the card in top end diagrams.

A reconsideration of the above facts will show that the horizontal lines *b c* and *a d* (Fig. 115) are drawn concurrently with the stroke of the engine piston—therefore the centre of the line *b c* (Fig. 115) would be reached by the pencil exactly as the engine piston was half-way towards the bottom of the cylinder; that is, the engine would then be at half stroke. Supposing at this point the influx of steam were suddenly stopped, the paper drum being impelled by the string attached to the cross-head would still continue to revolve, but the descent of the piston would allow the steam to expand, as greater space would be given to it in the cylinder, and the pressure would gradually decrease; the indicator pencil would therefore descend at the same time that the card revolved. The theoretical diagram formed in the case here supposed would be as in Fig. 117. Here *b c* is the steam line; at *c* the steam is cut off, and as it expands pressure decreases until the point *d* is reached; here the opening of the exhaust port causes the pressure suddenly to drop to its lowest point at *e*, and the same series of events would ensue as already described (See Fig. 115). The line *c d* is known as the expansion line or expansion curve. The straight line *f f*, drawn beneath the diagram, is called the atmospheric line, and is formed by bringing the pencil into contact with the paper whilst the drum is being moved by means of the cord, but before the steam had been admitted from the

engine to the indicator. This line will register on the diagram the pressure of the atmosphere, and will have been drawn whilst the indicator piston spring was in equilibrium, the indicator piston itself being neither pressed upwards by steam pressure, or extended downwards by vacuum. It will be obvious that all pressures registered above this line will be pressures above that of the atmosphere, and all such parts of a diagram as are below it are due to a pressure below that of the atmosphere, caused by the condensation of steam. It is only with condensing engines that the diagram extends below the atmospheric line. In diagrams from non-condensing engines, where the whole of the diagram is above the atmospheric line, the space between the lower line of the diagram and the atmospheric line represents the back pressure or resistance opposed to the movement of the piston by the exhaust steam in escaping from the cylinder.* In the present instance, spring No. 7 is supposed to be used, and the distance between the atmospheric line and line of counter-pressure

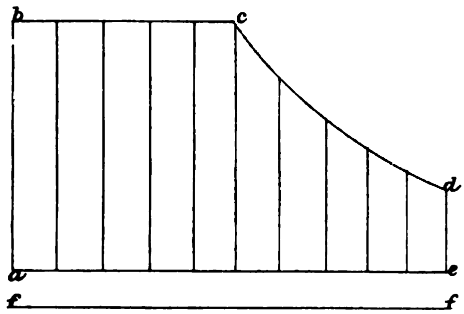


FIG. 117.

is $\frac{3}{8}$in. or $\frac{12}{32}$in., equivalent to 12lb. pressure, which would be the amount of back pressure on the stroke; and, as the counter pressure is exerted during the return stroke of the engine, whilst the part of the cylinder under observation is open to the exhaust, and the practical work of the engine is being performed by the steam on the other side of the piston, the 12lb. back pressure will neutralise an equal pressure of 12lb. on the steam side of the piston. The distance from the line *f f* to the line *b c* (Fig. 117) is 3in. = $\frac{96}{32}$in. = 96lb. pressure to square inch, which would be the pressure at which steam entered the cylinder, called the "initial pressure." It is seen, however, that 12lb. of this pressure remains in the cylinder to counteract an equivalent pressure on the other stroke; and therefore, for ascertaining the practical amount of work obtained from the steam on both strokes, it will obviously be necessary to deduct the

* This must not be confounded with the back pressure absolute, which is the pressure of the escaping steam plus the pressure of the atmosphere.

12lb. back pressure from the mean pressure of steam shown throughout the stroke. This we find already done for us on the diagram, for we have only to take our measurements from the line of counter-pressure instead of from the atmospheric line, and we get the real value of the work performed by the steam. It may be asked, then, "Of what use is the atmospheric line?" Simply for the purpose of ascertaining the power of an engine it is not absolutely necessary, for we have only to measure diagrams from each end of the piston in order to give sufficient data to make the necessary calculations upon; but in order to determine whether the steam is being advantageously used, and whether the engine is in proper order generally, the atmospheric line is of the highest consequence.

We will use the above ideal diagram (Fig. 117) for the purpose of illustrating the manner in which the indicated horse power is calculated. It must be remembered that the diagram is reduced to half size, so that, in making calculations, all measurements taken therefrom must be doubled. Divide the diagram into a given number of equal parts by means of the vertical lines or ordinates, as shown. In an ordinary diagram ten is the usual number of divisions; but where from any cause the outline is very irregular it is sometimes necessary to interpolate ordinates so as to get a greater number of measurements from which to deduce the average measurements. Measure the height of the diagram midway between each ordinate, beginning on the left. The first five measurements will be $2\frac{5}{8}$in.=84lb. pressure; the next ordinate will measure $2\frac{1}{4}$in.=72lb.; the next $1\frac{3}{4}$in.=56lb.; the following ordinate $1\frac{3}{8}$in.=44lb.; the next $1\frac{1}{8}$in.=36lb.; and the last $\frac{7}{8}$in.=28lb. The mean of these ten pressures will be 65·6lb., which will be the mean or average effective pressure of steam per square inch.*

The unit of power for the steam engine is an equivalent to one horse power, and it was determined by means of experiments made by James Watt that a strong dray horse would perform work equivalent to raising a weight of 33,000lb. one foot high per minute. To ascertain the power of any given engine, we find, by the method above described, the effective mean pressure recorded on the diagrams taken from each end of the cylinder; multiply this by the number of square inches contained in the area of the piston, and multiply the product so obtained by the number of feet per minute travelled by the piston. This will give the number of foot-pounds per minute exerted by the engine. If we divide this by 33,000,

* The theoretical mean pressure, according to Mariotte's law (which, however, steam would not obey in a perfect steam engine), would differ very slightly from this, but we have adopted the usual course followed in practice, in order to illustrate our meaning.

the quotient will be the indicated horse power. The formula for calculating the horse power of an engine is as follows: A=area of piston in square inches; P=mean pressure of steam on piston in pounds per square inch, S=speed of piston in feet per minute; I.H.P.$=\frac{\text{A P S}}{33{,}000}$.

For practical purposes it is necessary to take diagrams from both ends of the cylinder, in order to ascertain the pressure on both the upward and downward strokes of the engine; for it by no means follows that diagrams taken from different ends of the same cylinder will be alike. Bad setting of the side valve alone will result in a considerable difference in area of top and bottom diagrams. But in the present case we will suppose the diagrams from each side of the piston to be alike. If this cylinder were 20″ in diameter=314·1 square inches area, the number of revolutions were 100 per minute, and the length of stroke being 2ft., the piston speed would be 400ft. per minute. The calculation for I.H.P. would be as follows: $314{\cdot}1 \times 65{\cdot}6 \times 400 \div 33{,}000 = 249{\cdot}7$. The latter would be the I.H.P. exerted by the engine at the time the diagrams were taken.

The three diagrams, Fig. 118, originally published in *Engineering*, are from the engines of a once well-known Atlantic liner. These diagrams are selected for the purpose of illustration on account of their symmetry, for it will be seen they are not examples of the most modern practice. The engines are compound condensing, with three inverted cylinders. The high-pressure cylinder, having a diameter of 62in., is placed between two low-pressure cylinders, each 90in. in diameter—the length of stroke in each case being 5ft. 6in.

The scale of the diagram is marked at the side, the diagrams being here reduced to half the size of the originals. For the high-pressure diagrams, spring No. 8 was used; and for the low-pressure cards, spring No. 2. A diagram from each end of the cylinder is taken on each card in the usual manner. For the purpose of taking two diagrams on one card in this way the indicator is usually connected to the cylinder by means of a pipe leading into a T-piece, one branch connecting to the top and the other to the bottom end of the engine cylinder. The T-piece is generally formed by a two-way cock fitted at the junction of the pipes, so that the indicator can be made to receive steam alternately from either end of the cylinder by turning the cock. In the diagrams the line corresponding with 0 on the scale is the atmospheric line. On the high-pressure cards there is pretty constant back pressure of 20lb. per square inch above atmosphere, and this back pressure will represent the initial pressure in the low-pressure cylinders, less the loss from accidental causes, &c. In the diagrams taken from the low-pressure cylinders the atmospheric line will be seen near the

middle of the figure, and the work represented by the area below the line is due to the vacuum caused by the condensation of steam in the condenser. The line of perfect vacuum (not shown) would be near the figures 15 on the scales of the low-pressure cards, and the distance between that line and the lower lines of the diagrams would represent the absolute back pressures in the low-pressure cylinders—which in fact will be theoretically the total back pressure on the engines, as the back pressure in the high-pressure

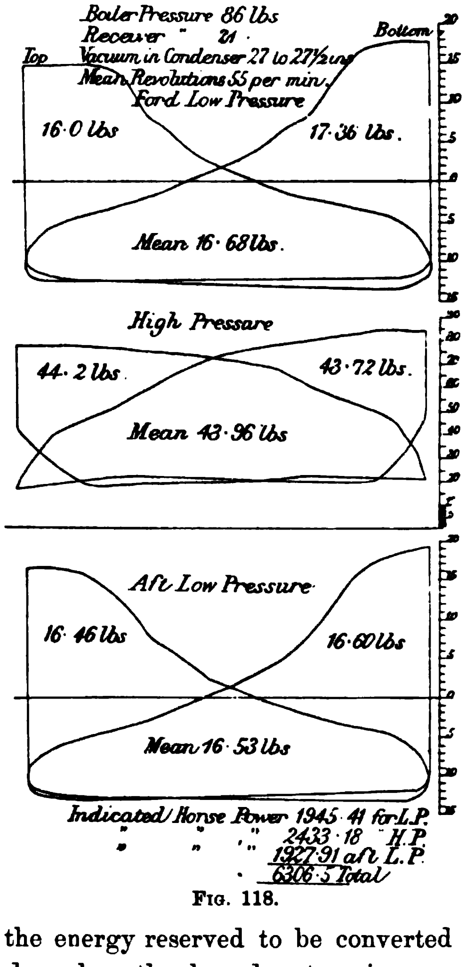


FIG. 118.

cylinder is simply the energy reserved to be converted into power in the low-pressure cylinders, less the loss due to minor causes. In the low-pressure diagrams, that part which is below the atmospheric line is caused by the extension downwards of the indicator spring; or, to use an expressive although not strictly accurate term, the indicator piston is "sucked" downwards by reason of the vacuum in the engines; and as the spring will be in equilibrium when the pencil is at the point 0 on the scale (in consequence of the indicator piston being then subject to an equal pressure on each

side), some force must be exerted to carry the piston below that point. This force is supplied by reason of the air being removed from, or rather being prevented from gaining access to, the interior of the cylinder on the exhaust side, and the steam pressure on the other side of the piston will therefore be working against a lesser retarding force than if the atmosphere had to be expelled. It will be obvious that there is an equal gain in the indicated power of the engine, whether additional force be exerted on one side of the piston, or a corresponding resistance to that force be removed from the other side; although, in considering the matter practically, the power absorbed by the air pump must not be forgotten. An equal force must be exerted in the tension as in the compression to a given distance of a spiral spring, therefore a given area of diagram below the atmospheric line is of equal value to the same area above the line.

It has already been stated that the indicator is useful, not only in affording a measure of the power exerted by an engine, but also for supplying data by which faults in construction of arrangement of the machinery can be ascertained; and it is in the application of such data that the engineer requires the use of his most critical powers of observation, and the aid of considerable practical experience carefully and industriously applied. We cannot hope here to do more than give a few examples or outlines of principles which may help to put the reader into the way of applying his own experience to the elucidation of such problems as may be brought before him. A yachtsman possessed of an indicator and the ability to use it should find no difficulty in acquiring the experience necessary to render him a good diagram reader. The number of steam yachts of all descriptions is so large, and there are so many owners who would be glad to get information about their craft and machinery from an unbiassed source, that there might soon be formed a collection of diagrams, taken from different engines working under different conditions, such as would enable the experimentalist, by a comparison of the data obtained with the known characteristics of the engines, to become a proficient on the subject. It is true that properly designed engines of a given type, and working under similar conditions, will give like diagrams; and it might therefore be considered that the cards of one set of yacht engines would probably be very similar to those of others. The intending experimentalist need not, however, fear any dull uniformity of excellence in the results obtained. In the majority of instances the diagrams taken from yacht engines will afford a sufficient field for the exercise of his critical acumen.

Indicator diagrams may become distorted, either from the indicator itself being out of order or improperly applied, or from faults in the

machinery operated upon. In order to obtain the best results the following points should be observed in fixing the indicator to the engine. The steam communication between the two should be as immediate and direct as possible; the most accurate diagrams are taken when the indicator is connected direct to the cylinder ends; but if connecting pipes must be used they should be ample in area, a less diameter than $\frac{1}{2}$in. being decidedly inadmissable, and in curves or bends $\frac{5}{8}$in. should be the smallest diameter of the pipe; all curves in pipes should be as easy and gradual as possible. It should also be seen that there are no points in the connections in which water from condensation could lodge. The pipes are sometimes clothed to keep them hot.

When it is inconvenient to attach the indicator, or the indicator connection, either to the top or bottom of the cylinder, a hole may be tapped in the side, care being taken that the piston does not cover the aperture when the engine is on centres; that is to say, the hole must lead into the clearance space, which is between the piston and cylinder covers when the engine is at either end of the stroke.

A want of proper attention to the adjustment of the cord by which the paper drum is actuated, may render the diagrams entirely worthless and misleading. If the part of the engine from which the motion is derived has a travel greater than 5in. (which is about the extreme length of a diagram that can be taken by the Richards' indicator), the motion must be reduced in some way. This is nearly always accomplished by one of the two following methods: By running the string over pulleys or wheels of different diameters, in which case the pulleys must be firmly fixed so as to allow no play at the instant of reversal of motion. Generally the most convenient manner of reducing the motion of the cord is by means of a radius bar, one end being attached to some convenient fixed point adjacent to the engine, the other being actuated by the engine crosshead, to which it is attached by means of a suitable link. Care must be taken that when the engine is at half stroke a straight line drawn through the centre of the extremities of the radius bar should be at right angles to the line of travel of the piston. Fig. 119 shows the arrangement described. A B is the radius bar, C being the link by which it is attached to the crosshead at the inner end, whilst the outer end is held by the bracket D, attached for the purpose to the engine framing. The distance of the point of attachment of the string from the outer end of the bar D must bear the same proportion to the total length of the bar that the travel of the cord bears to the stroke of the engine. Thus, should the latter be 18in., the length of the bar 30in., and the travel of the cord 5in.—the latter being, of course, the extent of motion of the paper drum—the string must be

attached to the radius bar 8·3in. from the fulcrum at B. Condensing engines working their pumps by a beam will present a ready means of

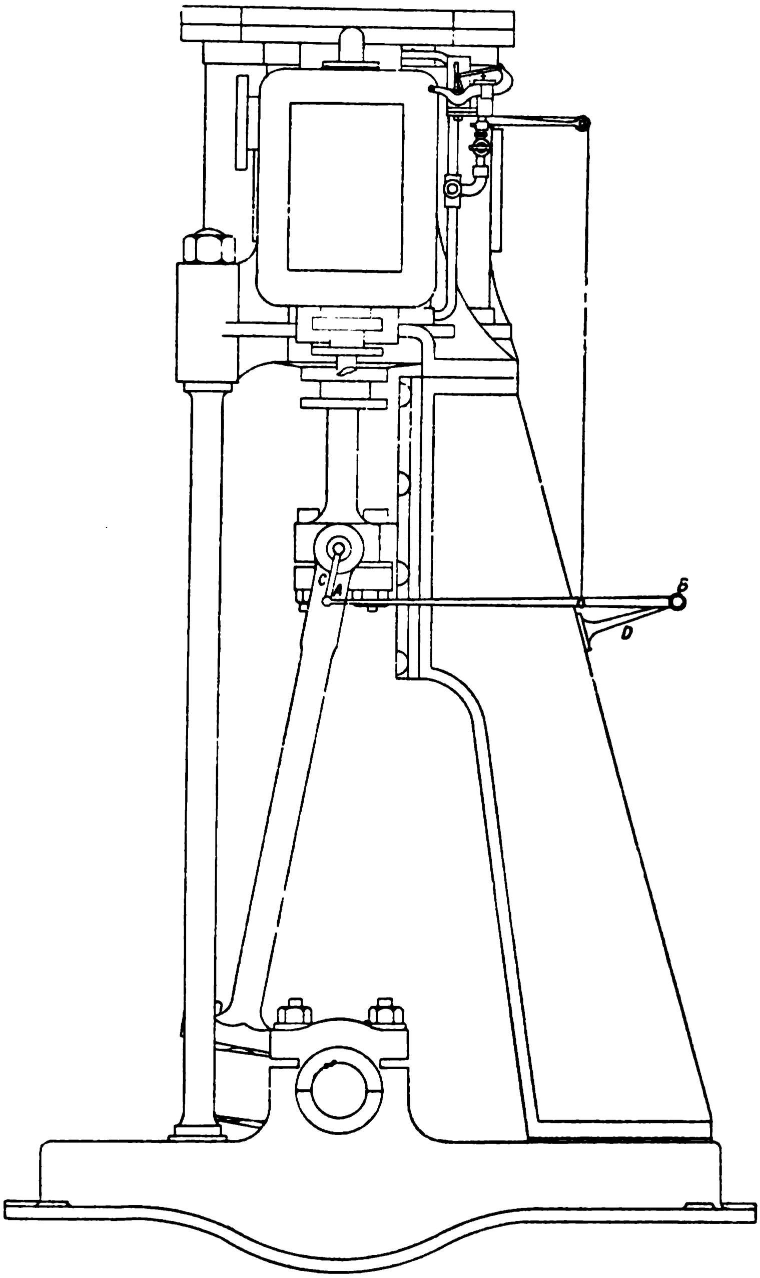


FIG. 119.

giving motion to the string in taking diagrams from the cylinder to which the beam is connected.

A point of importance to be observed is that the cord be not too new and unstretched. Should any considerable length of it be required, fine brass wire cord may be used for the purpose of obviating this difficulty. In quick-running engines more especially the cord may not convey a motion accurately corresponding to that of the engine. A good plan is to take a diagram at moderate speed, and compare the length of it with one taken at the higher speed. Should the latter be shorter in length, it may be concluded that it is incorrect, and must not be relied upon. Finally, in adjusting the cord, the operator must be careful that it is always in tension; that is to say, that the paper drum never is allowed to come against the stop placed in its interior for the purpose of arresting its motion, as in that case a part of the engine stroke would be performed with no corresponding movement of the drum. This would have the same effect as if the ends of the diagrams were cut off.

Either through impurities brought over by the steam or otherwise, the indicator piston and cylinder are liable to become damaged, or from some other cause the action of the instrument may not be so frictionless as is requisite for the purpose of getting accurate results. In order to test for this, when the indicator is in position on the engine, attach the string to the paper drum, but do not admit steam to the indicator. Press the piston down by applying the finger to the upper end of the piston rod, remove the pressure slowly and evenly, and draw the atmospheric line on the card in the usual way. Then reverse the operation by drawing the piston upwards, and pressing it with a slow and regular motion, afterwards applying the pencil to the card so as to again draw the atmospheric line. If there be no friction, the two lines will coincide, and only one line will be seen; but should there be any impediment to the free action of the instrument, the two lines will be separated a greater or less distance, according to the amount ot friction. To make a careful test of this nature, the pencil should be fairly sharp.

Indicator springs must be tested occasionally, as they are likely to lose part of their elasticity when long in work; and it must be borne in mind that, whilst in use, they are subjected to the heat of the steam. About 212° Fahr. is the temperature at which they are calculated to give the resistance corresponding to their respective scales. An allowance of about one pound in forty is necessary to be made when tested cold.

In studying a diagram taken from an engine using steam expansively it is absolutely necessary, in order to arrive at any definite conclusion as to the efficiency of the engine to construct the theoretical expansion curve which would represent the expansion of the steam under the conditions given, and to compare this curve with the actual expansion curve drawn

by the indicator pencil on the diagram. The law which governs the expansion of gases was discovered by Boyle in England and Mariotte in France about the same time. Although the elaborate experiments since made by the eminent French *savant* Regnault prove that saturated steam, such as is generally used—or, rather, ought to be generally used—in steam engines does not exactly follow this law; yet the difference is so small, that it is usually ignored by practical engineers, and Mariotte's law accepted as correct. This law is, that volumes of any given weight of gas at a given temperature are in an inverse proportion to the pressure to which the gas is subject, or, in other words, pressure varies inversely as the volume; so that if we expand, in a closed vessel, one cubic foot of steam at 100lb. pressure into two cubic feet of steam, the pressure will fall from 100lb. to 50lb.; or, again, if we have one cubic foot of steam at 100lb., and wish by compression to raise the pressure to 400lb., we must reduce the volume of the steam into one quarter of a cubic foot. In order to apply the theoretical curve of expansion to a diagram for practical purposes it is necessary that the clearance space between the piston and cylinder ends and the capacity of the steam passages be ascertained, and the area due to the steam contained in them added to the admission end of the diagram. Thus, if the port and clearance have a capacity equal to one-twelfth the space swept by the piston, it will be necessary to extend the diagram one-twelfth of its total length at the admission end. The necessity of this will be obvious, for if the steam be cut off at half the stroke of the piston—the area of the latter being one square foot, and the length of stroke 2ft.—there will be one cubic foot of steam in the cylinder at the point of cut-off, supposing we do not take into consideration the port and clearance; but, as the latter must necessarily be filled with steam before the piston can commence its stroke, we shall in this case have at the point of cut-off one cubic foot of steam plus the steam in port and clearance, the whole of which will expand to the point of release. Supposing, then, the initial pressure were 100lb., the terminal pressure, in place of being 50lb. (as it would be were there no port and clearance space), would be 50lb. plus the pressure due to the steam contained in port and clearance at the commencement of the stroke. In order to ascertain correctly the point of cut-off, the engine should be turned over by hand when the valve chest cover is off, and the point at which the valve closes the ports on both strokes carefully noted. In diagrams taken from ordinary marine engines the proportion of the stroke for which steam is admitted can only be approximately defined by means of the diagram alone, in consequence of the steam line gradually merging into the expansion curve, owing to the slow closing of the steam port and consequent gradual

fall of pressure. It must be remembered that the steam line does not end until the port closes, however much the line may fall previously.

Fig. 120 is a diagram, constructed for the purpose of illustrating the theoretical curve of expansion of steam as usually taken* in an engine with a piston one square foot in area and a stroke of two feet; steam at 50lb. pressure (above atmosphere) being admitted for half the stroke of the piston. The port and clearance may in this case be taken as equal to one-twelfth the space in the cylinder swept by the piston. For convenience this diagram is divided into twenty-six equal spaces or intervals, each one representing a volume of steam, twenty-four of these intervals represent the space swept by the piston, and the two others the port and clearance. The area A B F G is the port and clearance, and the rectangle, A C J G, is the steam at initial pressure above the piston at the point of cut-off.

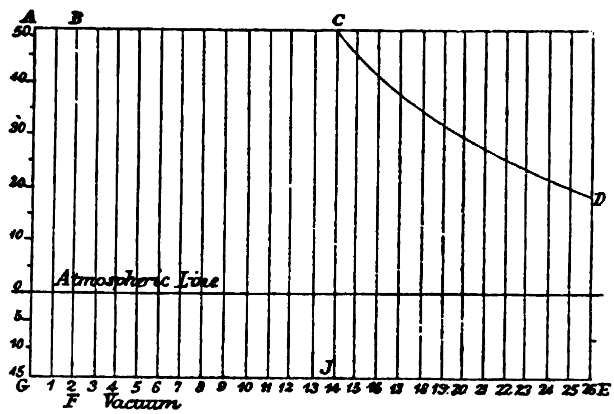


Fig. 120.

According to our custom in these elementary illustrations we do not take into account accidental disturbances in the working of the engine; for instance, our initial pressure is supposed to be maintained fully up to the point of cut-off, and the full vacuum is maintained from the point of release (which coincides with the end of the stroke) at 15lb. below the atmospheric line, the engine, of course, being a condensing engine. To construct the curve of expansion according to Mariotte's law, lay off the ordinates as shown, taking care that one of the ordinates coincide with the point of cut-off. Multiply the pressure of steam in pounds per square inch at the point of cut-off into the number of intervals up to that point, and divide the product so obtained by the number of spaces up to the point on the expansion curve at which it is required to find the pressure.

Following this rule we will construct the curve C D. The number

* The isothermal expansion of a gas would be the more correct expansion, as in a theoretical non-conducting cylinder steam should expand adibatically, and not follow Mariote's law.

of intervals up to the point of cut-off is 14, and the steam pressure 65lb. per square inch absolute. The pressure at ordinate 15 will be $65 \times 14 = 910 \div 15 = 60{\cdot}66$lb. The scale of our diagram being $\frac{1}{24}$in. = 1lb. pressure, the pressure 60·66, or roundly $60\frac{1}{2}$lb., will be represented by a distance of $2\frac{25}{48}$in. from the line of vacuum. Having marked off this distance on ordinate 15 we may proceed in the same manner to find the points at which the theoretical expansion curve intersects the remaining ordinates. The number 910 (initial pressure $\times$ volumes of steam at point of cut-off) now becomes a constant, and we have only to divide it by the remaining ordinates :

$910 \div 16 = 56{\cdot}87$	$910 \div 17 = 53{\cdot}52$	$910 \div 18 = 50{\cdot}55$
$910 \div 19 = 47{\cdot}89$	$910 \div 20 = 45{\cdot}50$	$910 \div 21 = 43{\cdot}33$
$910 \div 22 = 41{\cdot}36$	$910 \div 23 = 39{\cdot}56$	$910 \div 24 = 37{\cdot}71$
$910 \div 25 = 36{\cdot}40$	$910 \div 26 = 35{\cdot}00$	

Measure off from the vacuum line the distance due to the pressures on the ordinates, and draw the curve C D through the points so obtained which will give the theoretical line of expansion.

Fig. 121 illustrates a graphic method of constructing the hyperbolic

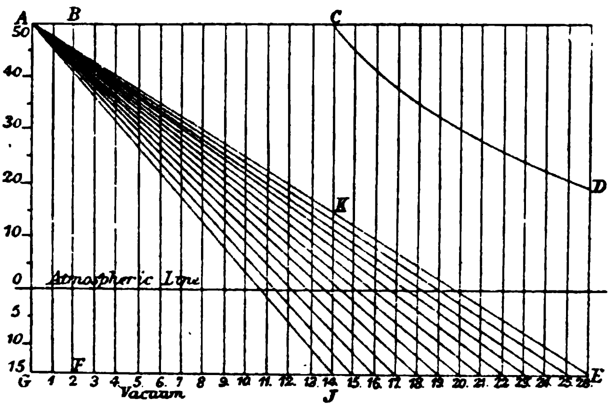


FIG. 121.

curve, and, as it obviates any necessity for calculation, and is very simple, it will probably be preferred in many instances.

Having found the point of cut-off, draw the ordinate C J from such point to the vacuum line. Divide up the remainder of the diagram to the point of release by means of ordinates into any number of parts. The port and clearance having been added (as in Fig. 120), from the point A to the foot of the ordinate, dividing the diagram at the point at which it is required to find the pressure, draw a straight line, and the distance from the point C to where this line cuts the ordinates C J will be a measure of the pressure at that ordinate to the foot of which the line is drawn. Thus,

supposing we wish to ascertain the pressure at ordinate 26 on the diagram, we should draw the line A E, and measure off the distance C K, which will equal the distance D E, the latter being a measure of the pressure of steam at the part of the stroke to which it corresponds.

There is still another method of applying the theoretical expansion curve which we will illustrate by means of Fig. 122, which is a diagram from the high pressure cylinder of the engine of a Clyde-built vessel. The cylinder is 39½in. in diameter and the stroke 3ft. 6in. The capacity of the port and clearance is ·0753 of the space swept by the piston. The length of the diagram being in the original 5in., we must add ⅜in. for port and clearance. Steam has been cut off early in the stroke by means of an expansion valve. This method, which we are about to

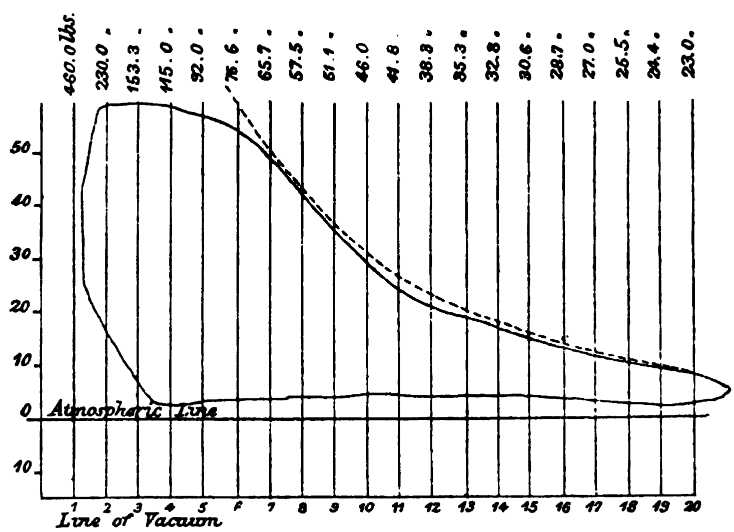


FIG. 122.

illustrate, of applying the theoretical expansion curve is useful in cases where the exact point of cut-off at the time the diagram was taken is not known. Select a point on the diagram near the termination of the stroke which shall be, as near as can be ascertained, the point at which the slide valve opens the exhaust port and allows the steam to escape to the condenser. The exhaust port here opens before the end of the stroke in consequence of the advance of the eccentric. Having divided the diagram, including the port and clearance space, into any equal number of intervals up to the point selected, draw in by hand the line of perfect vacuum. In the examples we have taken the vacuum is supposed to be 15lb. per square inch below the atmospheric pressure, but in cases where perfect accuracy is required the atmospheric pressure must be

ascertained by means of the barometer. Find, by measurement on the diagram, the absolute pressure at the point selected as the point of release, and multiply this into the number of intervals on the diagram, and the product will be the pressure of steam at the end of the first interval, according to Mariotte's law; by dividing the pressure thus found by the number of intervals on the diagram up to any point at which we wish to ascertain the pressure, we shall obtain the theoretical pressure at such a point. In Fig. 122 the pressure at ordinate 20 (which has been selected as the probable point of release, as beyond that the steam shows a more sudden decrease of pressure, undoubtedly caused by the opening of the port) is 23lb. absolute; and it must be remembered that in constructing the theoretical curve of expansion pressures absolute or pressures above perfect vacuum must invariably be taken. This pressure of 23lb. × 20 (the number of intervals) = 460, which would be the pressure in pounds at the end of the first interval, supposing the steam were compressed into a space in the cylinder represented by one interval on the diagram. Following the above rule, 460 ÷ 2 = 230, which would be the pressure at the end of interval No. 2, and in the same manner 460 divided consecutively by 3, 4, 5, 6, &c., would give the pressures 153lb., 115lb., 92lb., 76lb., &c., respectively, as shown on the diagram. We have now only to measure off these pressures according to the scale of the diagram, and through the points so obtained draw the theorectical curve, as shown by the dotted line. It has been said that the pressure at the end of space No. 1 would be 460lb., but the highest pressure registered by the diagram is only 74lb. absolute. As the reason of this discrepancy may not be quite obvious to those who are accustomed to the considera-tion of such problems as these, we will approach the question from another point of view. Supposing, then, the action of the engine to be reversed, and in place of steam being admitted at the left of the diagram, so as to impel the piston forward, that the steam be introduced at the point of release at the pressure there indicated—viz., 23lb. absolute—and the piston moved backwards by some extraneous means so as to compress the steam, thus causing the pressure to rise as the space occupied by the steam became smaller; the theoretical curve due to such compression, and consequent rise of pressure from a lower to higher pressure, would be the same as that due to a like amount of expansion, and would therefore be the dotted line already drawn on Fig. 122, and, other things being equal, the actual line of compression drawn by the indicator pencil would follow the actual line of expansion there shown. It will be seen that these two lines (the actual and theoretical expansion curves) agree pretty closely until the seventh interval is reached, and here the theorectical curve leaves the actual

line of the diagram. The reason is not far to seek, for the steam port opening would allow the additional pressure (caused by the supposed backward movement of the piston) to escape, and no higher pressure would be registered than that due to the initial pressure supplied from the boiler. If, however, the port could be kept closed, and the piston were still moved backwards, the two curves would ascend in nearly the same direction, and the 460lb. pressure at the end of the first interval would be attained, or nearly attained; supposing, of course, the means were at command to compress the steam to that extent, and there were no loss from other causes. It is not intended to be understood that that pressure would be actually attained, the case is merely put by way of illustration.

It should be stated that this diagram (Fig. 122) shows two faults; a loss of pressure both on the admission line and the steam line. This is shown by the rounding off of the top left hand corner of the diagram and the fall in the steam line, together with its gradual blending into the expansion curve.

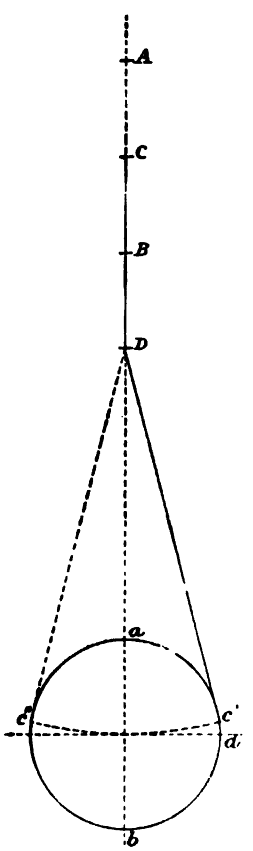


FIG. 123.

Reference has been made to the obliquity of movement of the connecting rod, and we will here say a few words in illustration of this subject. At first sight it might be taken for granted that when the main crank after starting from centres has completed one quarter of a revolution, that the piston would likewise have made exactly one half of its stroke; that is to say, with the crank starting at *a* in Fig. 123, the first quarter of a revolution would exactly coincide with half the stroke of the piston; but this is not by any means the case. In Fig. 123 A C B is the line of travel of the piston. The circle *a c d b c''* is the path of the crank. The length of the connecting rod is double that of the stroke. Supposing the piston to start at A it will when at C have completed half the distance it has to travel during one stroke. The point of connection between the piston and the connecting rods, or engine cross head, will then be at D, and consequently C D will be the piston rod. The position of the crank at one-quarter of a revolution will be *d*. If, however, we lay off from D the length of the connecting rod, it will only reach the point *c'*, and the latter point will therefore be the position of the crank with the

piston at half stroke. The shorter the connecting rod, compared to the stroke, the greater the obliquity of the former, and so much the greater will be the difference between the angle made by the crank when the piston is at half stroke and a right angle. Referring again to the figure, on the up stroke, the crank will have completed more than a quarter of a revolution when the piston is at half stroke, as shown by the dotted line D c'', which is equal to the length of the connecting rod. Further, as c' and c'' are the positions of the crank when the piston is at half stroke on the up and down strokes respectively, and as the length of the segment c', b, b, c'' is greater than that of the segment c'', a, c'; and as the crank travels at a constant speed, the piston will move with a greater velocity whilst performing the parts of its stroke corresponding to the movement of the crank c'', a, c', than during the part of its stroke corresponding to the crank movement c' b, c''; or, in other words, the piston makes the first half of its down stroke, and the second half of its up stroke in less time than the second half of the down stroke, and first half of the up stroke; or, simpler still, the piston travels more quickly in the upper half of the cylinder than in the lower half.

The almost universal description of engine placed in modern sea-going steam yachts is the inverted, direct-acting, compound, surface-condensing type, with two cylinders, or three cylinders, and in some cases four cylinders.

The history of improvement in marine engines is to a great extent a record of higher pressures, and the additional expansion thereby rendered possible. Steam can undoubtedly be expanded in a single cylinder as it can in two, or three, or more, supposing separate cut-off valves be used. In the compound engine there is an actual loss caused by the expansion of steam in passing from one cylinder to another, during which time it does no work. This is what is known as the "drop" between the indicator diagrams taken from the respective cylinders, and it is undeniably a loss that does not occur with simple engines in which the expansion takes place continuously in one cylinder only. The loss, however, is not so great as would at first sight appear by the comparison of the diagrams, as the steam, supposing no water to be present, becomes superheated during expansion in the intermediate receiver (a consequence of no external work being done), and this excess of heat above that due to the pressure, is to some extent utilised when the steam passes to the low pressure cylinder. No action of this sort occurs with air or perfect gases; steam, however, if dry, does become superheated; but, as a matter of fact, there is generally some water at the end of the stroke, which would boil off and become steam. If the water were sufficient it would, of course, prevent

superheating, but the water boiled off into steam would contribute its share of work. In order to form a just estimate of the loss from the "drop," the theoretical curve of expansion through the total range of expansion should be drawn, instead of the difference between the terminal pressure in the high and the initial pressure in the low pressure cylinder alone being considered. The method of constructing this diagram was laid down by Professor Rankine, and has since been copied into several text books treating of the steam engine.

If engine cylinders could be made of a perfectly non-conducting substance, then no doubt the simple expansive engine would be, perhaps, on the whole the preferable type, but it is not rash to predict that such a substance, suitable for making cylinders, will never be discovered, and the compound engine of two or more stages is likely to hold its own so long as steam is used as a motive power; indeed, as the pressure and consequently the heat of steam increases, "compounding" will be carried still further. Already the triple compound engine, with a high, medium, and low pressure cylinder, has established its superiority over the two-cylinder compound, and there are, working with considerable success, several types of quadruple compound engines having four cylinders, the steam expanding from one to another consecutively.

In order to explain this part of our subject more fully it is necessary that we should give a few details of the laws which govern the relation of temperature and pressure of steam towards each other. Saturated steam at any given pressure has a corresponding temperature, and any heat taken from the steam will result in its partial or complete liquefaction, and any heat subsequently added to the steam will cause a portion or the whole of that which has liquefied to again become steam. If saturated steam be contained in a closed vessel in which there is no water, any heat imparted to it will cause what is called superheating, but superheating cannot take place so long as water is present with the steam, as the heat goes to evaporate the water, and thereby raise the pressure, for the higher the pressure the higher the temperature. Saturated steam, therefore, is steam which is at the temperature due to its pressure; any abstraction of heat will cause liquefaction, and any addition of heat will lead to superheating if no water be present. Thus steam in an ordinary boiler at 50lb. pressure absolute has a temperature of 281° Fahr. If all communications with the boiler were closed, and the fire continued in operation, more water would be evaporated, and the pressure and temperature raised until the whole became steam. Supposing this occurred when a pressure of 500lb. to the square inch were reached, the corresponding temperature would be 467° Fahr., and any further addition of heat would

go to superheat the steam, whilst any abstraction of heat would cause water to again form.

When saturated steam, at a given pressure, first passes into the cylinder a part of the heat is abstracted from the steam by the colder metal composing the cylinder, and a portion of the steam is liquefied, forming water. Steam from the boiler presses on to supply the place of that liquefied, and this goes on until the cylinder walls are of the same temperature as the steam, or the slide valve shuts off the supply. When the latter occurs the steam goes on expanding, parting with its heat in doing work, and it thus becomes lower in temperature than the metal of the cylinder. The latter in turn gives up the heat obtained from the steam when first admitted, and the water in the cylinder resulting from the first liquefaction becomes again evaporated, and adds to the pressure driving the piston forward. It will thus be seen that the walls of the cylinder rob the steam on its first admission, and give up what was abstracted during the latter part of the stroke. This would be very well (at least very well in comparison to what actually occurs) if the action were perfect; and, indeed, in one respect would be advantageous as tending to equalise the turning moment on the crank, but, unfortunately, the greater part of the heat retained by the cylinder metal is not given off until the return stroke, when the exhaust port is open and the cylinder in communication with the condenser—a part of the machinery where heat is not only not necessary, but absolutely injurious, as tending to increase the volume of refrigerating water required. The back pressure is also increased to an appreciable degree by the escaping steam.

It will be easily seen from what has been said how desirable it is, from one point of view, to have as small a range of temperature as possible in the cylinder. Heat, however, is equivalent to work, and the whole power of the steam engine is measured by the range of temperature between the initial and final pressures. But if we must have a wide range of temperature we need not confine it to one cylinder. If we cut the expansion short in one cyclinder, and finish it in another, in neither will there be the same difference of temperature that would exist if the expansion were carried on in one cylinder. The high-pressure cylinder will not be at any time so cold, nor the low-pressure cylinder at any time so hot, whilst the heat absorbed by re-evaporation that takes place on the return stroke of the high pressure cylinder will not go directly to the condenser.

It is a good thing to work steam expansively, but, like many other good things, it may be overdone; and the same may be said of an extension of the compound system. So long as steam pressures did not rise above 30lb., the simple expansive engine was a very efficient apparatus,

and the long time that elapsed after this type was invented* before it came into use was no doubt due to the difficulty of making boilers to stand heavy pressures. With the general introduction of the cylindrical boiler, however, the chance for the compound engine arose, and its use soon became all but universal in marine practice. The use of steel plates for boiler shells, and the introduction of Fox's corrugated furnaces enabled pressures to be increased from 60lb. to 100lb. and then on to 160lb.; since then sea-going steamers and yachts have been constructed with boilers pressed from 180lb. to 220lb. At the time of writing details have just been received of a cargo steamer with engines designed by Mr. Mudd, the cylindrical boilers of which are pressed to 255lb. to the square inch. These increases in pressure have enabled triple compound and quadruple compound engines to supersede the two-cylinder type in the same way that the latter ousted the simple expansion engine.

The first triple compound engine was fitted to the steam yacht Isa, in 1877, and owned by Mr. Hugh Andrews, of the Royal Thames Yacht Club. The engine was made by Messrs. Douglas and Grant, of Kirkcaldy, for a working pressure of 120lb., and may be regarded as the precursor of the many triple cylinder engines now in use.

Isa is a two-masted yacht, 118ft. length of keel; 18ft. 9in. in extreme breadth; and 10ft. 5in. in depth; her yacht tonnage being 248 tons. The high-pressure cyclinder was placed above the intermediate cylinder, the respective diameters of the three being 10in., 15in., and 28in., and all with a 2ft. stroke, driving on to two cranks; one boiler, 8ft. 9in. diameter, and 8ft. 6in. long, with two furnaces, and 106 2¾in. tubes. The shell plates of iron, 1in. thick; boiler proved to 250lb., and for a daily working pressure of 120lb., indicating a little over 200-horse power, on a consumption, it is said, of 300lb. of coal per hour.

Since the success of the triple engine has been established it has practically been universally introduced in steam yachting.

The economy of using steam at very high pressures is primarily due to the fact that for all pressures the consumption of heat, or, in other words, fuel, to produce it, is approximately the same. Next, as the efficiency of high pressure steam is much greater than that of comparatively low pressure, and as it can be obtained at a nominal extra cost, there is every inducement to use it. For instance, the total heat expended in evaporating 1lb. of water from 100° Fahrenheit to 312°, equal to 80lb. pressure, is 1108·6 thermal units; and the total heat required to evaporate 1lb. of water from 100° to 358°, equal to 150lb. pressure, is 1122·4. That is, the expenditure

* The compound engine was invented by Hornblower in 1781, and some time after Woolf added the condenser.

of heat to produce steam of 150lb. pressure is only 1·0117 times greater than that required to produce steam at 80lb. pressure. In other words (leaving out of sight the greater efficiency of steam at 150lb. pressure), if 500lb. of coal were consumed in maintaining 250 I.H.P. for one hour from steam at 80lb. pressure, only 506lb. of coal would be required to maintain the same I.H.P. from steam at 150lb. pressure. But, as a matter of fact, owing to the rapid increase in the efficiency of the steam at the higher pressures, such a considerable reduction would be made in the weight of steam employed to develope the 250 I.H.P. that the consumption of coal would be considerably less than 500lb. So far as present experience goes, the actual saving of fuel in using steam of 150lb. pressure with three cylinders, as against steam of 80lb. pressure with two cylinders, is 20 per cent.; and 18 per cent. between steam of 100lb. pressure and 150lb. pressure. This takes into account a slight loss in the efficiency of the triple engine due to an increase of friction from the extra number of parts, &c. Thus, to follow out the foregoing illustration, only 420lb. of coal would be required to produce the 250 I.H.P., as the consumption per I.H.P. per hour would be 1·6lb. instead of 2lb. This greater economy of the higher pressure steam is mainly due to the higher rate of expansion possible, representing work done; to the smaller range of temperature in the cylinders; and to the reduced "drop" in the receiver pressure from the terminal pressure of the steam as it leaves the high pressure cylinder. But also a measure of the economy is attributed to the initial stress being smaller than that consequent upon the heavier load on the high pressure cylinder in the two cylinder engine; and to the possibility of getting comparatively a greater amount of work out of the low pressure cylinder. The loads on the three cranks being more distributed in the triple compound engine than they are in the double, it follows that the working stresses are smaller; in fact, it has been calculated that the average stresses are 40 per cent. smaller. The running of a three crank shaft is also more even than that of a two crank, and consequently the efficiency of the propellor is greater; also the three crank engine can be run at lower speeds than a two crank, which may be of occasional advantage. It will be noted that these advantages will not exist if the triple cylinder engine is arranged with the high pressure cylinder above the intermediate pressure cylinder, making what is known as the tandem engine, as there will be only the two cranks as in the ordinary two cylinder compound engine; but the other advantages due to the expansion of the steam and the small range of temperature in each cylinder will be the same whether the triple engine be tandem or not. Of course, the objection to three cranks is that the engine must take up more room in a fore and aft direction,

and hence the object of the tandem arrangement. This question of space is the principal objection which can now be raised to the three crank triple compound engine, as it is termed. Of course, there is the question of the greater heat due to steam of 150lb. pressure, but that could scarcely be urged as a valid objection to using steam of 150lb.; there are also the questions of the extra care required in using the boiler, which might mean a better class and more highly paid engineer; and there is the question of the greatly increased first cost of the triple compound engine. If a yacht owner contemplates long voyages the reduced coal bill will soon make up for the extra cost of the boiler and machinery, and for general use the reduced wear and tear will go a long way towards compensating for the extra first cost. With regard to the greater fore and aft space required by the triple cylinder engine, it may be said that the latest design and arrangement of valve chests show that the triple cylinder engine need occupy so little more room in a fore and aft direction, that the objection under this head practically disappears; and, moreover, in some cases the reduced consumption of fuel might admit of a cross coal bunker being dispensed with, and thus there would be an actual gain in fore and aft engine-room, boiler and coal space.

By the use of a steam jacket to the cylinder supplied direct from the boiler, it was expected at one time that the evils arising from liquefaction in the cylinders would be got over. Experience in practical work has not completely justified this assumption. By experiments it has been found that the feed water used per indicated horse power per hour when the jackets were not in use was 16·87lb., and when the jackets were filled with steam it was 17lb.; on another trial the figures were 16·97lb. and 17·16lb. But, even allowing a high efficiency for steam jackets when properly attended to, it is seldom that that desirable condition can be insured at sea.

Mr. Thornycroft with his small fast launches first brought home to the marine engineering world the value of high piston speeds. His well-known and now historical Miranda ran with a speed of piston amounting to 800ft. per minute; and some of the torpedo boats since built have gone as high as 880ft. per minute—a speed up to that time never seen in any general practice excepting that of locomotive driving. Although the number of reciprocations for any given speed of piston are naturally greater with small than with large engines, yet it is less dangerous to run the light working parts of a launch engine at a high speed than those of the machinery in larger vessels. Considerable advances have, however, been made in this respect within the last year or two; and the twin screw engines of the cruisers of the

Phaeton class (5500 I.H.P.) ran at 800ft. per minute, the stroke being 4ft. These are horizontal engines; but the Atlantic liner Umbria on her trial trip made seventy revolutions per minute, the piston speed being 840ft., the horse power developed being 14,300 indicated. This piston speed is said to have been exceeded in some of the more recent vessels; the Majestic making, we believe, 900ft. per minute, whilst the battle-ship Renown reached to a still higher piston speed on her full speed trial with one set of engines.

With high piston speeds the variations in temperature of the steam have not so much time to communicate themselves to the metal of the cylinder, and the process of liquefaction and re-evaporation which has been previously described is therefore less marked. With high piston speeds steam jacketing would be less effective, as the evil it is intended to counteract is not present to the same degree.

A quick running engine is *ceteris paribus* a light engine, for naturally two revolutions will do more towards propelling a boat than one. It may be said generally that high piston speed is one of the most necessary features in fast vessels. Lightness in the moving parts is a prime necessity with increased velocity of movement, and the application of steel to marine engineering practice has contributed largely to this end. In the navy cylinder liners, pistons, piston rods, connecting rods, and shafting are made of steel, which not only strengthens and lightens the parts, but, by its superior hardness and smoothness of surface, reduces the wear on the rubbing parts. Cast steel dished pistons are now very much used in the highest class of marine engines, and are said to afford a saving of 35 per cent. in weight over the old cast iron pistons. Accurate counter-balancing of the moving parts is also a most important feature in the design of quick running engines.

The Institution of Mechanical Engineers appointed a "Research Committee on Marine Engine Trials." Up to the present time this committee has issued two reports, which, even if they do not bear out the hopes that were entertained when the committee was formed, are valuable contributions to our too limited knowledge on this subject. The reports have taken the form of papers read by Professor Alexander Kennedy, F.R.S., chairman of the committee, at meetings of the Institution.

On pages 251 and 252 is given, in the form of a table, the results of the trials made by this committee so far as they have been made public.

The data given in the table is so full that the figures speak for themselves. A few remarks in further explanation may, however, be made.

The Meteor is a passenger vessel running between London and Edinburgh. Such vessels are not built so much with regard to economy

in coal consumption as to obtain speed, and we know the two qualities are largely antagonistic. This must be remembered when comparing the performance of this ship with the others, especially the second triple compound ship, the Tartar. It will be seen that the efficiency of the engine in the Meteor is high, and, although the efficiency of the boiler is less than that of the Fusi Yama, the combined efficiency is the best shown. It is to be regretted, however, that the water condensed in the jacket was not measured separately as it is doubtful how far the jacket was effective. It would have been interesting had a trial been run with the jackets entirely shut off. Of course such points as these detract from the value of the figures given; but, in the case of marine engine trials, we must be thankful for what we can get and use our brains in putting the data to practical use. With regard to the Tartar, the priming of the boiler was so excessive that many of the figures depending immediately on boiler performance must be taken with considerable reserve. It is for this reason that the efficiency of the engine is not stated.

The Fusi Yama may be described as an ordinary trading vessel working under the usual conditions. She is 632 registered tonnage. The Colchester is one of the Great Eastern Railway Company's passenger boats. She has twin screws and runs between Harwich and Antwerp. The Tartar is described as supplying "an excellent example of modern economical engines in a cargo-carrying steamer." Her gross registered tonnage is 2389 tons. When she was tried by the committee she had only the water ballast in her tanks. The weather was very bad during the run, which was from the Thames to Portland, and there was often considerable difficulty in taking observations.

The inverted compound condensing engine of two or more stages has been so universally adopted in marine practice, and its merits above all other types are so generally acknowledged, that reference to other descriptions is hardly necessary in connection with sea-going screw steamers. Variations in designs are sometimes adopted. The cylinders are occasionally placed tandemwise one above the other, or diagonally in the vessel, both these plans being followed in order to save fore-and-aft space, but the judicious engineer adheres to the usual arrangement. At times, however, the exigencies of design and construction necessitate the introduction of unusual arrangements when the vessel is intended for any special service. We shall not here attempt, for obvious reasons, to give directions for designing marine engines. Those who wish for fuller directions in this respect would do well to study the many excellent works devoted to the science; although engine designing is to be but imperfectly learned from books, the only satisfactory school being the drawing office, the shop, and the steamship at sea.

COMPARATIVE RESULTS OF THE TRIALS OF FOUR STEAMERS "METEOR," "FUSI YAMA," "COLCHESTER," "TARTAR."

1	Name of vessel	"Meteor."	"Fusi Yama."	"Colchester."	"Tartar."
2	Date of trial	June 24, 1888	Nov. 14 & 15, 1888	Nov. 9, 1889	Nov. 27, 1889
3	Duration of trial ... hours	17·15	13·95	10·88	10·08
4	Type of engines	Triple, 3 crank jacketted at sides	Compound, not jacketted	Twin Compound, not jacketted	*Triple, 3 crank jacketted
5	Cylinder diameter, high-pressure ... inches	29·87	27·35	(two) 30	26·03
6	" " intermediate ... "	44·03	...	...	42·03
7	" " low-pressure ... "	70·12	50·3	(two) 57	68·95
8	Stroke, length ... "	47·94	33	36	42
9	Boilers, number of main boilers (ordinary return tube)	2	1	2	2
10	" single-ended or double-ended	Double	Single	Double	Double
11	Furnaces, total number	12	3	12	8
12	Heating surface, total ... square feet	6648	2257	5820	5226
13	" " tubes ... "	5760	1689	4770	4366
14	Grate area ... "	208	52	220	161
15	Total heating surface to grate area ... ratio	32·0	43·4	26·5	32·5
16	Tube surface to grate area ... "	27·7	32·5	21·7	27·1
17	Grate area to flue area through tubes ... "	...	4·05	5·51	4·50
18	" " area through funnel ... "	5·04	3·21	4·77	4·19
		lb. per square inch.	lb. per square inch.	lb. per square inch.	lb. per square inch.
19	Mean boiler pressure above atmosphere	145·2	56·84	80·5	143·6
20	" admission pressure high-pressure cylinder above atmosphere	134·4	50·3	{ 64·3 { 59·4	121·4
21	" effective pressure high-pressure cylinder	58·46	30·74	{ 45·65 { 42·07	36·89
22	" " " intermediate cylinder	19·50	...	...	20·07
23	" " " low-pressure cylinder	12·38	10·87	{ 13·42 { 12·42	7·18
24	" " " total reduced to low-pressure cylinder	29·9	19·9	24·8	19·8
25	" exhaust pressure low-pressure cylinder below atmosphere	11·6	10·9	{ 10·6 { 10·5	10·5
26	" vacuum in condenser below atmosphere	12·17	12·48	12·49	12·9
27	Revolutions per minute, mean	71·78	55·59	{ 86·0 { 87·1	70·0
28	Indicated horse-power, mean total	1994	371·3	{ 1022·5 { 957·2	1087·4

* The high-pressure cylinder jacket was shut off during trial.

COMPARATIVE RESULTS OF THE TRIALS OF FOUR STEAMERS "METEOR," "FUSI YAMA," "COLCHESTER," "TARTAR."—*Continued.*

	Name of vessel	"Meteor."	"Fusi Yama"	"Colchester."	"Tartar."
29	Coal burnt per minute ......lb.	66·75	16·45	95·7	32·0
30	,, ,, hour ...... ,,	4005	987	5742	1920
31	,, ,, square foot of grate per hour ...... ,,	19·25	18·98	26·1	11·93
32	,, ,, ,, ,, total heating surface per hour ...... ,,	0·602	0·437	0·987	0·367
33	,, ,, indicated horse-power per hour ...... ,,	2·01	2·66	2·90	1·77
34	Carbon value of 1lb. of coal as used ...... ,,	0·878	0·878	0·913	1·031
35	,, equivalent per indicated horse-power per hour ...... ,,	1·76	2·33	2·65	1·82
36	Feed water per minute ...... ,,	497·7	131	717	359·4
37	,, ,, hour ...... ,,	29,860	7860	43,020	21,564
38	,, ,, square foot of total heating surface per hour ...... ,,	4·49	3·48	7·39	4·13
39	,, ,, pound of coal ...... ,,	7·46	7·96	7·49	[11·23]
40	,, ,, ,, ,, from and at 212 deg. Fahr. ...... ,,	8·21	8·87	8·53	[13·06]
41	,, ,, ,, carbon value from and at 212 deg. Fahr. ...... ,,	9·62	10·10	9·34	[12·67]
42	,, ,, indicated horse-power per hour ...... ,,	14·98	21·17	21·73	[19·83]
43	Calorific value of 1lb. of coal as used ......thermal units	12,770	12,760	13,280	14,995
44	Percentage of line 43 taken up by feed water	62·0	67·2	62·0	
45	,, ,, carried away by furnace gases	21·9	23·5	28·0	22·1
46	,, ,, lost by imperfect combustion	3·6	0·0	1·3	0·0
47	,, ,, expended in evaporating moisture in coal	1·2	0·9	0·4	0·0
48	,, ,, unaccounted for	11·3	8·4	8·3	
49	Heat taken up by feed water per minute ......thermal units	528,600	141,100	788,700	[403,600]
50	,, turned into work per minute ...... ,, ,,	85,240	15,870	84,630	46,490
51	,, taken up by feed water per indicated horse-power per minute, thermal units	265·6	380	398·4	[371·2]
52	Efficiency of boiler (line 44) ......per cent.	62·0	67·2	62·0	
53	,, ,, engine (line 50 + line 49) ...... ,,	16·1	11·2	10·7	[11·5]
54	,, ,, ,, and boiler combined (line 52 × line 53) ...... ,,	10·0	7·6	6·6	9·7
55	Mean velocity of steam through water surface in boilers per minute...feet	...	6·28	8·6	3·43
56	Space occupied by boilers per indicated horse power......cubic feet	2·72	4·53	2·52	4·33
57	Weight of engines, boilers, &c., with water, per indicated horse power, tons	0·20	0·27	0·20	0·27
58	Clearance surface, high-pressure cylinder ......square feet	...	...	{ 22·6 { 22·6	
59	,, ,, intermediate ,, ... ,,	—	—	—	—
60	,, ,, low-pressure ,, ... ,,	...	...	{ 56·9 { 56·9	
61	Speed of vessel, mean, during trial ......knots	14·6	...	14·4	

Having dealt with the question of triple compounding, there is not much left to say on the various types of machinery. In large steamships two low-pressure cylinders are often used, steam from the high-pressure cylinder exhausting into each one independently of the other—a plan that must not be confused with the triple compound, in which the steam passes from the high to the intermediate, and then to the low-pressure cylinder. The reason that two low-pressure cylinders are used is mainly on account of the difficulty in getting such large castings as would be required were one cylinder used. This trouble seldom arises in connection with yacht machinery. The Lady Torfrida has, however, two low-pressure cylinders arranged in the way described. In the smaller classes of steam yachts,

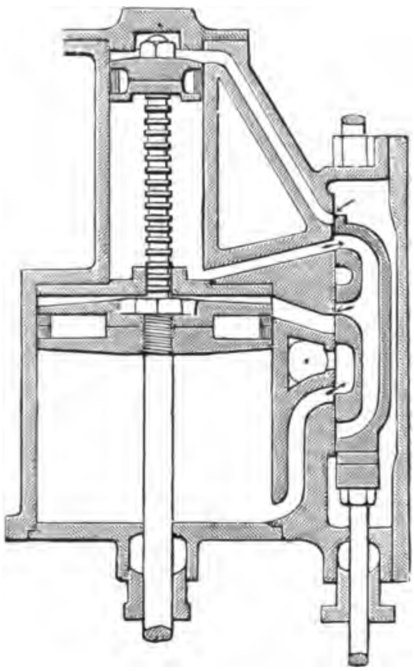


FIG. 124.

several novel types of engine have been from time to time introduced. Of late the plan of splitting the low-pressure cylinder has been followed in marine engines not of the largest size. Several of the torpedo boat destroyers have engines on this plan.

Another ingenious launch engine, the invention of Mr. Kingdon, Torquay, has been introduced by Messrs. Simpson and Strickland, of Dartmouth, and is worthy of notice here, as it has been largely used for launches and small steam yachts. It is an inverted direct-acting tandem engine, and has now been in use some years. As shown by Fig. 124, the steam is distributed to both the high and low pressure cylinder by a single

valve, there being one valve chest common to both cylinders. The full boiler pressure, less the small loss between the boiler and the engine, is therefore pressing on the whole valve area of the engine, forcing it up to the cylinder face, and the loss from this cause must be considerable. So far as we are aware the makers have never adopted any means for relieving this abnormally large valve from the full boiler pressure. The high-pressure cylinder is placed immediately above the low-pressure cylinder, the bottom of the former being also the top of the latter. There are, therefore, no stuffing boxes between, as in ordinary tandem engines, but for keeping the steam from passing from one cylinder to the other reliance is placed on a number of concentric grooves which are turned in the rod connecting the two pistons, this rod being a good sliding fit to the hole in the division between the two cylinders in which it works. When the rod is travelling upwards, the under part of the high-pressure piston is subject to pressure whilst the top end of the low-pressure cylinder is open to exhaust; during that time it is of course undesirable for steam to pass through the hole in which the piston rod works, and it is claimed that any steam attempting to pass is caught in the grooves in the rod and brought back by the upward travel. On the down stroke, when the steam is exhausting from the under side of the top piston to above the lower piston, it is of course no disadvantage for the steam to pass. With this engine steam is carried almost full stroke, the expansion being obtained by the difference between the capacity of the two cylinders. Messrs. Simpson and Strickland adopt the outside pipe condenser, of the type to which reference has been made elsewhere in these pages. They have obtained very successful results by its use. It is most effective when its external surface is kept clean, and a coat of paint renders it almost inoperative. They have also a special form of feed and air pump which works admirably, being, so far as we have ever been able to hear, practically unfailing in its action. For launches the Kingdon has come into extensive use; also larger engines for yachts have in some cases given satisfaction.

An engineer in the Royal Navy made some experiments with a set of these engines in the summer of 1884 in a 26ft. launch. The cylinders were two 2in. and two 5in., with 4½in. stroke. The following were the results:

DATA.	RESULTS.
Number of hours run, 6.	Speed = 6·95 knots per hour.
Miles run, 48.	Water evaporated = 29·8lb. per I.H.P.
Average steam pressure, 65lb.	Water evaporated = 10·04lb. per pound of coal.
Water evaporated, 125½ gallons = 1255lb.	Coal burnt = 20lb. per hour.
Coal burnt, 120lb.	Coal burnt = 2·85lb. per I.H.P. per hour.
Grate surface, 3·7 square feet.	Coal burnt = 2·5lb. per mile.
Indicated horse power, 7.	Coal burnt = 5·4lb. per square foot of grate per hour.
Revolutions, mean, 350.	
Vacuum, 25in.	

It is often thought that economy in the matter of steam, or in other words in coal, is not looked for in steam launches; this perhaps may be true in some cases, where a launch is only required for an occasional run of an hour, but where the run comes to nine or ten hours it is quite a different affair, and in selecting engines for a steam launch the conditions of running should always be considered.

Next to the introduction of the triple compound engine, the subject that claims most attention from users of steam in the present day is the application of forced draught for urging the combustion in the boiler furnaces. Increasing the steam generative capacity of a boiler by means of the steam blast in the chimney is the common method of obtaining higher power than ordinarily required. The use of the steam jet is, however, highly objectionable in many ways, the principal, in a pleasure craft, being the hissing of the escaping steam—a most aggravating sound, especially to the engineer, as it is generally associated in his mind with badly designed machinery in which the boiler is not up to its work. The use of the steam blast also causes irregularity in the feed by abstracting water from the boiler which is not returned, and for the destructive effect it has on the funnel. By means of the steam jet, an increased combustion of 40 to 50 per cent. may be obtained over that due to natural draught, but the gain in power is not above 15 per cent., showing a loss in economy of from 25 to 35 per cent. By substituting a centrifugal fan or blower, by which air is forced under the grate bars and through the burning fuel, the evils accompanying the steam blast are obviated, whilst nearly all the advantages gained by the use of the latter are obtained in a far greater degree.

There are several ways of applying forced draught by the fan, but the one that may be said to have come into general practical use in this country is by blowing air into the stoke-hold, which is made substantially air-tight for the purpose, so that all the air forced in by the fan must pass into the ash pit, and thence through the fire and up the chimney. This is the plan adopted in torpedo boats, and the enormous power contained in small compass and within a low weight of machinery, and the consequent high speeds reached by these vessels are mainly due to the use of the fan.

Shutting up the stoke-hold so as to render it air-tight may be, to some extent, objectionable, more especially where the engine and stoking space are undivided. To meet this difficulty the plan of supplying the ashpit with air-tight doors and blowing air directly into it has been tried successfully in the case of the steam yacht Speedy by using Parson's turbine system. There is one small drawback to this, for as a higher air pressure is obtained in the furnace than in the stoke-hold, there will be an escape of hot gases and flame so soon as the furnace door is opened. Another

difficulty that attended the first introduction of this system was the unequal distribution on the surface of the grate of the air supplied, as one part of the bars might be receiving the full force of the blast, whilst another would be comparatively uninfluenced by it.

Plans have been proposed for overcoming the difficulty of the escaping flames by shutting off the draught before opening the furnace door; but accidents have occurred through neglect of the precaution. The late Mr. Willans introduced an excellent plan for forcing the ash pit draught in small boats, which overcomes this difficulty.

Blowers or fans have also been fitted into the base of chimneys of steam boilers in order to draw the air through the fire, notably in the case of Martin's "induced draught" in the steam yacht Fauvette. It has been suggested that a satisfactory forced draught might be obtained by injecting air under pressure into the base of the chimney in much the same way that the ordinary steam jet is applied. The French naval authorities are said to have tried this method in a war vessel by way of experiment. Although this plan may have advantages, we do not anticipate it is likely to come into use for yachting purposes.

Trials have been made with a torpedo boat to ascertain if means of creating sufficient blast other than by the closed stoke-hold system could be introduced, and the fan was driven so hard as to give an air pressure of 12in. of water. Nozzles of different sizes, from 2in. up to 5in. in diameter, and a slit ring placed in the chimney, but the results were disappointing, the best obtained being with a nozzle 3½in. diameter, but this was no better than the ordinary draught, and was not one-half so effective as a draught equal to 2in. of water applied in the ordinary way with closed stoke-hold.

On the whole, then, it would seem that the way of applying the fan, which has proved itself to be most successful (and which was patented by Mr. L. E. Fletcher, M.I.C.E., about thirty years ago) is by means of keeping the stoke-hold under pressure, but it is not approved of for yachts.

The French Government have used fan draught with ordinary boilers for several years, whilst Messrs. Sir W. G. Armstrong and Co. have carried the system out successfully in some fast cruisers which they have built for foreign Governments. Provision is now made for the application of a system by closed stoke-holds in all the most important ships in the Royal Navy.

Recently forced draught has fallen considerably in public opinion, and one distinguished naval officer has gone so far as to style it "the invention of the Evil One." Those, however, who take public opinion—even when fortified by the support of gallant admirals—as their guide in engineering matters, are likely to go far astray. The accidents to Her Majesty's ships,

which have been so picturesquely dealt with by reporters of the daily press during the naval manœuvres, as far as they have been caused by forced draught at all, have been caused by the abuse of forced draught, and have nothing to do with the principle.

The best way to supply the necessary air for combustion in a marine furnace is by mechanical means. During the annual meeting of the Institution of Naval Architects held in 1883, Mr. R. J. Butler read a paper on the subject of forced draught, which was based on some trials of two of Her Majesty's ships, the Satellite and Conqueror. In the 1886 session of the same institution the late Mr. R. Sennett, formerly Engineer-in-Chief to the Royal Navy, also read a paper on this subject. Mr. Howden has also read papers before this institution, in which he dealt with his system of forced draught. These papers form a most important contribution to the literature of this subject, and as the facts illustrate fundamental principles we cannot do better than quote them, although the experiments are now some years old.

The Satellite is a single-screw ship of 1420 tons displacement. Her I.H.P. on trial with natural draught was 1116. She has compound condensing engines, with high-pressure cylinder 36in. diameter, and low-pressure cylinder 62in. diameter, length of stroke being 30in. The cooling surface in the condenser is 1900ft. The boilers, four in number, are of the long circular or gunboat type, previously described, in which a large combustion chamber with a hanging bridge is situated at the back of the furnace, beyond which again extend the tubes. The total grate area is 110 square feet, the total heating surface being 2920 square feet, 2430 square feet of this being comprised in the tubes, which are of iron. The funnel is 50ft. high, measured from the furnace dead-plate, the cross area being 15 square feet. For forcing the draught there are two fans, each 5ft. in diameter, actuated by a single-cylinder engine 7in. in diameter by 4in. stroke.

This vessel was tried against two sister ships, the hulls and machinery being alike in all respects excepting that the Satellite had her fans and machinery for forcing the draught, appliances which the Heroine and Hyacinth, the other two vessels, lacked. The results of these trials are given in table A.

From column 10, line 5, in this table it will be seen that the use of the fan gave an air pressure sufficient to raise a column of water varying between $1\frac{1}{4}$in. to 2in., and this resulted in each square foot of grate area becoming about 63 times per cent. more effective for steam generating purposes than when working with natural draught, the latter being shown in column 1. Further, the fan draught renders a given area of grate

TABLE A.

	Natural Draught Trials.			Steam Blast Trials.		Satellite Forced Draught Trials.				
	1 Satellite.	2 Heroine.	3 Hyacinth.	4 Heroine.	5 Hyacinth.	6	7	8	9	10
1. Date of trial	April 3, '82	May 30, '82	July 25, '82	May 31, '82	July 25, '82	May 10, '82	July 5, '82	July 11, '82	July 11, '82	July 11, '82
2. Number of boilers used	4	4	4	2	4	2	4	3	3	3
3. Duration of trial in hours	4	6	3	3	2	3	½	½	½	2
4. Number and diameter of blast nozzles	...	...	...	two $\frac{9}{16}$in.	four $\frac{9}{16}$in.	...	...	...	...	...
5. Air pressure in inches of water	...	...	...	...	...	1 to 1½	¼	½	1	1¼ to 2
6. Mean steam pressure in engine room	...	...	...	...	...	79	90	86·5	78·7	80·05
7. Mean steam pressure in boilers in pounds	84	82·6	81·6	83·9	85·0	...	...	...	...	...
8. Mean vacuum in inches	26·5	26·1	26·0	26	25·1	26·6	25	25·38	23·88	23·41
9. Mean revolutions per minute	98·52	104·2	105·8	89·4	111·6	95·4	113·5	103·36	110·66	121·45
10. Mean pressure in pounds: High pressure cylinder	32·36	32·07	31·6	22·45	34·05	27·52	29·95	29·94	33·7	31·93
12. Mean pressure in pounds: Low pressure cylinder	13·85	12·9	14·1	9·63	16·9	10·58	12·52	12·85	14·9	14·45
13. I.H.P. High pressure cylinder	493	515	515	310	586	405	524	469	575	598
14. I.H.P. Low pressure cylinder	623	612	680	392	859	459	647	605	751	799
15. Total	1116	1127	1195	702	1445	864	1171	1074	1326	1397
16. Area of fire-grate in square feet	110	110	110	55	110	55	110	82·5	82·5	82·5
17. I.H.P. per square foot of grate	10·15	10 25	10·87	12·76	13·1	15·7	10·6	13·0	16·0	16·9
18. Tube surface of boilers per I.H.P. in square feet	2·18	2·16	23·0	1·73	1·68	1·41	2·08	1·70	1·37	1·30
19. Total heating surface per I.H.P. in square feet	2·61	2·59	20·4	2.08	2·02	1·69	2·5	2·04	1·65	1·56
20. Area of funnel in square feet	15	15	15	*7·5	15	15	15	15	15	15
21. Maximum temperature in boiler rooms	111°	108°	98°	98°	100°	86°	...	...	...	...
22 Mean temperature in boiler rooms	90°	87°	84°	87°	80°	76°	...	...	...	...

* Diaphragm was placed in funnel.

surface over 29 per cent. more effective than when the steam blast is used, as shown by column 5 and 10 respectively.

It will be noticed that the total power developed was not always greatest with the highest pressure, the reason of this being that the engines would not take all the steam generated by all the boilers when working with the higher pressures for draught, and therefore some of the boilers had to be thrown out of use. On this account line 17, in which the power developed per square foot of grate is given, should be the one used in forming an estimate of the varying powers obtained under the different conditions of trial enumerated.

One attempt was made to run all the boilers using the fan draught and commencing with an air pressure of 1¼in., but the engines would not take the steam, so the trial was abandoned after 1570 I.H.P. had been reached, the engines making 126 revolutions.

It is interesting to note that ¼in. air pressure with stoke-hold closed gives about the same power as when running with natural draught in the ordinary way, as may be seen by comparing columns 1 and 10. With ½in. air pressure the result was about the same with regard to power developed as that obtained with the four $\frac{6}{16}$in. blast nozzles in operation, the I.H.P. per square foot of grate being respectively 13·1 and 13·0 as shown in columns 5 and 8.

The 5ft. fans of the Satellite when working at 400 revolutions per minute sustained an air pressure equal to 1½in. of water, and each additional fifty revolutions produced approximately an increase of 0·3in.

In feeding the boilers care was required to prevent the water from rising higher than about 6½in. over the crowns of the fire boxes, as slight priming occurred whenever that height was exceeded. During the Conqueror's trials, however, there was no symptom of priming. The boilers were in each case filled with fresh water for the purposes of the trials. The best Welsh coal was used, and the stokers were trained men from the reserves.

During the trial of the Satellite, detailed in column 6, the temperature of the escaping gases in the uptake was as high as from 1000 to 1200 Fahr. as taken by pyrometer. On the trial of the Heroine (column 4) a pyrometer similarly placed recorded a temperature from 775 to 850 Fahr.

Mr. Butler estimated from the data he had collected that, with suitable engines, the steaming power of the low boilers may be, by employing forced draught, increased by about 30 per cent. beyond the maximum power hitherto obtained with the steam blast, and this increase of effect is even greater in the case of the high (return tube) boilers.

With regard to the endurance of the boilers, there can be no doubt that the frequent use of the forced draught would produce a great diminution in the life of those parts subjected to the intense heat. But probably, under the conditions that will obtain, this will not be of serious moment. An examination of the boilers of the Satellite and Conqueror after the trials, showed they had not suffered to an unusual extent by the exposure to the intense heat for the short time the trials lasted. Only two or three of the iron tubes were found to be weeping in the low boilers; and in the Conqueror's a few seams and rivets, and about twenty of the tubes, were leaking slightly.

It is much to be regretted that no trials were made with these ships to ascertain the effect of forced draught on the consumption of coal. Naturally, however, the ratio of fuel to the power developed advanced at a very rapid rate. It will not be supposed, however, that forced draught is in itself uneconomical; in fact, the reverse of this is true. The reason the higher air pressures were less economical in the Satellite was, that her boilers were designed to work with natural draught, and therefore there was not heating surface enough to absorb the additional heat generated when the fan was in use, with a fair approach to an economical result.

The late Mr. Sennett, in his paper, dealt with the forced draught trials of five war vessels, and his information is, so far, more interesting than Mr. Butler's, because he gives figures as to coal consumption. We have not space to deal with the trials of all the vessels described by Mr. Sennett, but will merely give the details of the trial of the Caroline, as she is a sister ship to the Satellite, the machinery being practically the same in both vessels.

Forced Draught Trial of H.M.S. "Caroline."

Date of trial		March, 1885.
Duration of trial		6 hours
Number of boilers used		2
Mean steam pressure in boilers		84·52
Mean air pressure in boiler room in inches of water		1·5
Mean pressure in cylinders in pounds per square inch	High pressure	43·9
	Low pressure	12·79
Mean revolutions per minute		77·8
Mean piston speed in feet per minute		389
I.H.P.		983
Area of fire grate in square feet		54·5
I.H.P. per square foot of fire grate		18·02
Heating surface per I.H.P. in square feet	Tubes	1·24
	Total	1·43
Coal used per I.H.P. per hour in pounds		2·54
Coal used per hour in tons		1·11

Speaking fairly within ordinary limits and in general terms, the power

of a steam boiler may be said to be governed by the amount of coal it is capable of burning in its furnace, and therefore the quantity of heat generated; whilst economy may be taken to depend on the sufficiency of area of heating surface provided for the absorption of the heat. Taking, then, boilers of any one type, and burning the same description of coal under similar conditions of draught, &c., the power of the boiler would be expressed by the area of a fire-bar surface. To illustrate this we will suppose a boiler with 100ft. of grate surface, and, burning, say, 20lb. of coal on each square foot per hour—to each square foot of grate we will allow 30ft. of heating surface—the consumption per hour would be 0·66lb. of coal per square foot of heating surface, and this proportion would allow a sufficient amount of heating surface to absorb the heat liberated by the combustion of the amount of coal stated, with a fair approach to economy. From an economy point of view alone the heating surface can be hardly excessive, but, naturally, there are practical considerations involved which limit the extension of a marine boiler to a point short of the highest economy obtainable.

In a well-designed marine boiler the temperature of the escaping gases should not be above 600°, whilst as low a temperature as 500° may be reached. We will take the latter figure for the sake of comparison, and allow the furnace temperature to be 2500°. Taking these temperatures and the above rate of combustion and proportions of grate to heating surface, we find that, putting aside other details that might arise, there has been sufficient heating surface to absorb four-fifths of the heat generated, the remaining fifth escaping up the chimney; but, nevertheless, doing the necessary work of creating the draught if no artificial means are used. Supposing, now, the stoke-hold were closed, and a fan blower were started, so that the increased draught caused double the amount of fuel, that is 40lb., to be burnt per square foot of grate per hour. In that case the furnace temperature would be greatly increased, and the transmission of heat to the water in the boiler would be more rapid than with the lower temperature due to the natural draught, and consequently far more steam would be generated.

The rate of transmission of heat may be taken to vary as the square root of the difference in temperature. With the quicker draught, a larger volume of heated gases would pass the same area of heating surface in a given time (as we have assumed the quantity of coal burnt to be doubled, we should have 1·13lb. per square foot of heating surface per hour), and therefore would not be in contact with the heating surface sufficiently long to be reduced to a temperature low enough to produce a fair economy, that is to say, too much heat would be going up the chimney. In the Satellite's

trials, when steaming with 1in. to 1½in. fan-draught, the temperature of escaping gases was 1200°. Assuming the chimney temperature should be as high as 600° for natural draught, we find that the amount of heat allowed to escape up the chimney in the former instance was double that which it would be in the latter. In cases where the fan is used the loss is more to be deplored, as there is no necessity for any heat in the funnel to create a draught; and were it possible within reasonable limits of size and weight of boiler, it would be desirable, so far as economy is concerned, to deliver the products of combustion to the funnel at a temperature no higher than that due to the steam generated.

In this case we have supposed the ratio of grate to heating surface differs materially from that of the Satellite's boiler. It would not be possible, from the data now at our disposal, to make any reliable calculation of the loss in coal economy due to the increased draught used on the various trials made with the Satellite; we have the experience of the Caroline, as given by Mr. Sennett; and Table B. has been compiled from the result of some trials made with a torpedo boat's boiler at Portsmouth to show the loss in economy and gain in power due to forced draught. The total heating surface was 618 square feet, and the grate surface 18·4 square feet.

TABLE B.

TRIALS OF TORPEDO BOAT'S BOILER (LOCOMOTIVE TYPE).

Duration of experiment	2h.	2h. 7min.	1h. 39min.	1h. 27min.
Air press in stoke-hold	2in.	3in.	4in.	6in.
Revolutions of fan per minute	575	665	818	986
Temperature of feed-water in degrees Fahr.	53·5	57	54	56
Temperature of funnel	1073	1192	1260	1444
Coal consumed per hour	925lb.	1177lb.	1472lb.	1815lb.
Coal consumed per hour per square foot of fire-grate	49lb.	62lb.	78lb.	96lb.
Coal consumed per hour per square foot of heating surface	1·5lb.	1·9lb.	2·38lb.	2·93lb.
Water evaporated per hour	6530lb.	7770lb.	9320lb.	10,840lb.
Water evaporated per pound of coal	7·06lb.	6·6lb.	6·33lb.	5·97lb.
Evaporation per pound of coal, reduced to equivalent at 212° from 100° Fahr.	7·61lb.	7·08lb.	6·81lb.	6·41lb.
Evaporation per hour per square foot of grate	345	411	493	573
Evaporation per hour per square foot of heating surface	10·8lb.	12·9	15·5	18·0

Coal used, "Nixon's" navigation, with 9 per cent. of ash.
Steam pressure during trials, between 115lb. and 117lb. above atmosphere.

The proportion of heating surface to grate surface (33·5 to 1) is ample for ordinary rates of combustion, and with such rates one would anticipate a good result in fuel economy, as the coal then burnt would not be above 0·4lb. to 0·6lb. per square foot of heating surface per hour, representing an amount of heat generated which it would be within the power of the total

heating surface to absorb, so as to give an economical result. If, however, we take the highest rate of combustion in Table B.—viz., 96lb. per square foot of grate—we find that 2·86lb. of coal was consumed per square foot of heating surface, an amount far in excess of general engineering practice, and a very poor economy would result. This is borne out by Table C. nearly 4lb. of coal are required for each H.P. developed, even when running considerably below top speed.

A torpedo boat is an extreme example of the enormous power which can be obtained with light weight and in small compass of machinery by use of fan draught, and also of the price that must be paid for it. To illustrate this more fully, Table C has been prepared, in which the performance of a torpedo boat is contrasted to that of two yachts. The machinery in one developed about the same power as that in the torpedo boat, whilst the other is a fine-lined vessel of somewhat the same displacement as the torpedo boat. The great disparity in other details shows the length to which the influence of forced draught may be carried. The details of the Oriental's performance are taken from trials conducted by Mr. John Inglis, jun., whose high reputation is a sufficient guarantee for their accuracy. The Leila is an American yacht, and was made the subject of a very elaborate series of trials by the U.S. naval authorities.

TABLE C.

	First-class Torpedo Boat.	Leila, a Composite Yacht.	Steam Yacht Oriental.
HULL.			
1. Length over all	......	100ft.	143ft.
2. Length on L.W L	88·3	95ft. 3in.	120ft. 3¼in.
3. Maximum breadth	10·6	15ft. 4in.	20ft. 1¼in.
4. Displacement	32·5 tons	37·27	266
Co-efficient of fineness	......	0·3721	0·46
MACHINERY.			
5. Fore and aft space occupied	13ft.	......	27 t. 6in.
6. Description of engines	Compound condensing	Compound condensing	Compound condensing
7. Dimensions of engines	12¾ and 20¾ × 12	9 and 16 × 18	18 and 32 × 24
8. Fore and aft spaces occupied by engines	6ft.	5ft. 6in.	
9. Description of boiler	Locomotive type	Herreshoff coil	Return tube
10. Total heating surface	618ft.	529ft.	696ft.
11. Total grate	18·4ft.	25·96	30ft.
12. Ratio	1 to 33·58	1 to 18·68	1 to 23·20
13. Diameter of boiler	......	......	10ft.
14. Fore and aft space occupied by	13ft.	7ft.	8ft. 6in.
15. Propeller, number of blades, diameter and pitch	Three 4ft. 6in. × 6ft	Four 4ft. 7in. × 8ft.	7ft. × 10ft.
16. Total weight of engines, propeller, &c.*	7 tons †	......	22 tons 16 cwt.
17. Total weight of boiler and water	7 tons †	4 tons 16 cwts.‡	27 tons 4 cwt.
18. Total weight of all machinery	14 tons	......	50 tons.

* Including torpedo gear. † Estimated by constructor.

‡ Calculated without chimney.

TABLE C.—*Continued.*

	Torpedo Boat.		Leila, Composite Yacht.		Oriental.
	Full speed trial.	Lower power trial	Ordinary running.	Trial at lower power.	
PERFORMANCE.					
19. Mean boiler pressure	133·8	...	129·4lb.	104·5	80lb.
20. Vacuum	23·8	...	25·83in.	25·92	28in.
21. Mean number of revolutions	443	...	221 ⅌ min.	192 ⅌ min.	146
22. Mean piston speed	886ft.	...	663ft. ⅌ min.	576ft. ⅌ min.	584ft. ⅌ min.
23. I.H.P.	469	340	149·94	99·90	330
24. I.H.P. per ton displacement	14·43	...	4·023	2·680	1·250
25. Co-efficient of performance $\frac{V^3 \times D^{\frac{2}{3}}}{P}$	224	...	178	...	185
26. Weight of engines, &c., per I.H.P. in pounds	33·43	46·00	...	...	151·7
27. Weight of boiler and water per I.H.P. pounds	33·43	46·00	71·70lb.	107·52	184·8
28. Weight of all machinery per I.H.P. in pounds	66·86	92·00	...	...	338lb.
29. Heating surface per I.H.P. in sq. ft.	1·31	1·817	3·52	5·29	2·10
30. Grate surface per I.H.P.	0·029	0·0541	0·173	0·259	0 091
31. Coal burnt per hour	...	1332lb.	332lb.§	215lb.§	660lb.
32. Coal burnt per I.H.P. per hour	...	3·92lb.	2·01lb.	1·72lb.	2lb.
33. Coal burnt per foot grate per hour	...	72·4lb.	12·7lb.	8·2lb.	22lb.
34. Coal burnt per foot heating surface per hour	...	2·155lb.	0·68lb.	0·44lb.	0·94lb.
35. Air pressure in stoke-hold	5½in.	...	Nat. draught	Nat. draught	Nat. draught
36. Speed in knots	21·75	18·75	13·45	11·74	11·4
37. Duration of trial	6 knots.	...	15 knots.	87 knots.	—

§ Coals used on these trials very bad, containing 15·09 per cent. of incombustible on first trial, and on second trial 20·47 per cent. On the torpedo boat trial, "Nixon's" Navigation was used, which contains about 4 to 5 per cent. of incombustible. The actual consumption has been reduced to an equivalent of 5 per cent. of incombustible.

Supposing, for the sake of comparison, that the vessels quoted are the best that could be produced of their respective classes, an examination of the table shows that, should it be considered desirable by the owner of a vessel similar to the Oriental to travel at the rate of 18¾ knots per hour without materially increasing the power, he would have to reduce the displacement from 266 tons to about 32 tons; but as the machinery in the Oriental weighs 50 tons, some means would have to be devised for reducing this weight (given we retain the same machinery), and the only plan at present known to engineers would be by the use of forced draught. The adoption of this system would lead to the consumption of over double as much coal per unit of power as that now burnt in the yacht; so that, roundly speaking, the new high-speed vessel of small displacement would consume as much coal in covering a distance of 18¾ knots as would serve for moving the 266 tons of the existing Oriental 22¾ knots. On the sacrifice in accommodation that would have to be paid for the higher speed there is no need to comment. Yachtsmen are generally fully alive to the importance of this factor in the design of their vessels. The question of coal endurance is also one of great importance at these high speeds. Mr.

Thornycroft says that one of his first-class torpedo boats ran from London to Cherbourg at the rate of 11 knots per hour and burnt 2½ tons of coal; whereas the torpedo boat mentioned in the second column of Table C would consume that quantity of fuel in about three hours after having covered 56¼ knots when steaming at 18¾ knots per hour, that is, supposing 4lb. of coal were burnt per I.H.P. Mr. Yarrow has stated that one of his 100ft. torpedo boats of 40 tons displacement will steam 100 miles at the rate of 10 knots an hour with a total consumption of 1 ton of coal. It is unnecessary to produce further evidence to prove that it would be impossible on our present knowledge of engineering to construct a vessel which might be fairly called a sea-going yacht to steam 18¾ knots with engines of the same power as those in the Oriental. It must not, however, be taken as contained in this proposition that no vessel of greater displacement than that of the first-class torpedo boat in question could be driven at the given speed on the same power. This vessel is simply taken as the exponent of the best existing type of vessel for very high speed.

The plan of taking the power to be exerted by the engines as the fixed quantity in designing a vessel is not usual; the general manner of proceeding being for the dimensions of the vessel and speed required to be laid down, and the machinery then made sufficiently powerful for the purpose. Unfortunately our present knowledge of the behaviour of vessels of the class of the Oriental at speeds so high as 18¾ knots is not sufficient to enable us to predict with any degree of accuracy what would be the power required at that speed. Within recent years there have been some very high speed vessels of larger size built and tried. The Wiborg and El Destructor are a pair of twin-screw torpedo cruisers, constructed for foreign governments by J. and G. Thomson, of Clydebank; whilst the Boxer is one of the torpedo boat destroyers, built for the British Navy by Messrs. Thornycroft and Co. The following particulars of these vessels will be of interest:

	H.M.S. "Boxer."	"Wiborg."	"El Destructor."
Length between perpendiculars	200ft.	142ft. 6in.	192ft. 6in.
Breadth	19ft. to 6in.	17ft.	25ft.
Depth at centre	13ft.	9ft. 6in.	13ft.
Draught due to normal displacement	—	4ft. 7in.	6ft. 3in.
Draught due to load displacement	7ft.	5ft. 2in.	7ft.
Diameter of cylinders	19in., 27in., & 2 of 27in.	14in. & 24in.	18½in., 27in., & 42in.
Stroke	16in.	15in.	21in.
Number of boilers	3	2	4
Pressure of steam	210lb.	130lb.	145lb.
Grate surface in square feet	180 sq. ft.	56	144
Normal coal	70 tons.	14 tons.	37 tons.
Radius of action due to ditto at 10 knot speed	2500 knots.	1400 knots.	2050 knots.

With the two foreign vessels five trials were made, of which the following are given by the builders as the results:

	"Wiborg."			"El Destructor."	
	1st trial.	2nd trial.	3rd trial.	1st trial.	2nd trial.
Load on trial	41	37	70	88	88
Displacement at trial	138	131	167	385	385
Mean speed for 3 hours	19·96	20·6	18·55	22·56	22·68
Revolutions	381	380·1	362	292	292·3
I.H.P.	1303	1405	—	3784	3829

Running at the easy speed of 11·6 knots, the mean I.H.P. being 297, the El Destructor burnt 1·95lb. of coal per I.H.P. per hour. At full speed, with an air pressure of 2¼in. of water, there was burnt 2·4lb. of coal per I.H.P. per hour. The displacement loaded of the Boxer was 250 tons, and the speed on the official trial was 29·175 knots maintained for three hours, the I.H.P. being 4,800.

It is not intended, of course, to put these vessels forward as examples of what could be done in steam yachting. Young gentlemen of ample means sometimes express a wish to build a steam yacht to beat everything that has gone before. A few hours' running at full speed in such a craft as El Destructor would be likely to assuage such vain desires. The speed in this vessel was only obtained by sacrificing nearly every feature of constructive solidity. At the time she was delivered to her purchasers fears were expressed that the very light scantling might turn out too light for actual use, and we never heard anything to dissapate these fears. Certain it is the vibration when running at speed might be described by a much stronger word than unpleasant. It should be added, however, that the Destructor ran from Falmouth to Muros, in Spain, a distance of 495 knots, in 24 hours. In regard to the Boxer, which is of course a much later vessel, improvements in the balancing of engines and other matters have so reduced the amount of vibration in high speed craft, that there is much more comfort on board. Still, there is the very large space occupied by machinery, and other points to be taken into consideration, and these render such vessels quite unfit for pleasure purposes.

Some of the first-class torpedo boats approach sea-going yachts in size, although they differ from the latter in general characteristics. The Ariete, built by Thornycroft and Co., is a twin-screw vessel of this class. She is 147ft. 6in. long and 14ft. 6in. wide. Her load draught, however, is only 5ft., a light draught being aimed at in the design. A novel point is that the boat has two rudders; in fact, the whole

design at the after-part is peculiar. The engines are of the ordinary two-cylinder compound surface-condensing type, the cylinders being 14½in. and 24½in. diameter by 15in. stroke. Steam is generated by two Thornycroft pipe boilers, before described. On a six-run trial in the Lower Hope this vessel steamed at the rate of just over 26 knots per hour. The steam pressure was 151lb. to 153lb., the air pressure for forced draught from 3½in. to 3⅝in., and the revolutions from 388 to 398 per minute. On a two hours' run the speed was just under 25 knots per hour. The collective horse power of the two pairs of engines at full speed is said to be 1550 indicated.

The torpedo boat, with locomotive type of boiler, is hardly designed to run with natural draught, and therefore does not afford a good opportunity for ascertaining the actual gain in speed by applying forced draught on any one vessel. We have, however, records of three trials made at Portsmouth with two small vessels having water-tube boilers, which were run both with forced and natural draught. These boats in many respects are not unlike the big, so-called steam launches one sees in the yachting season; and, as a high result in speed was obtained with them, and, moreover, as the record comes on trustworthy authority, somewhat full particulars of the boats and machinery are now given :

Hull—wood, two skins, ⅜in. and ¼in., one diagonal one fore and aft; Michigan pine, white oak, timbers, 1⅜in. square, spaced 12in., decked throughout with mahogany, movable steel hatches; open well aft, to hold twelve people, with water-tight floor, fitted with self-emptying valves at sides; cabin abaft engine space; bunker capacity, 2 tons.
Length on deck, 48ft.; on L.W.L. 46ft.
Breadth on deck, 9ft.; on L.W.L. 7·5ft.
Depth, exclusive of keel, 5ft.
Displacement on trial draught, 7·44 tons.
Co-efficient of fineness, 0·396.
Wetted surface, 355·5 square feet.
Engines, inverted, compound surface-condensing, with air and feed pumps, worked by side levers; cylinders, 8in. and 14in. diameter, by 9in. stroke; blowing engine, single cylinder, 2½in. diameter, by 5in. stroke, working centrifugal fan direct; diameter of fan, 42in.; propeller, four blades, 3ft. diameter by 4ft. 1in. pitch.

	Trial A.	Trial B.	Trial C.
Duration of trial	6 knots	Round I. of W.	10 hours.
Force of draught in inches of water	2·66	Slight draught used occasionally...	Nil.
Mean boiler pressure	145lb	93lb	*53·35.
Mean vacuum	20·29in.	24in.	22in.
Mean revolutions	453 per min.	333 per min.	273 per min.
I.H.P.	No cards ...	76·62	36·7.
Speed	15·124 knots	10·97 knots	8·635 knots.
Force of wind	1		4 to 5.
Sea	Smooth		Rough.

* Engines linked up.

Some trials at full speed over longer distances were made with two sister vessels to the above, the best result bring as follows:

	First run over base.	Second run over base.	Third run over base.	Fourth run over base.	Mean of four runs.
Steam pressure	132lb.	130lb.	130lb.	126lb.	129lb.
Revolutions	478 per min.	471 per min.	454 per min.	460 per min.	466 per min.
Revolutions of fan engine	667 per min.	667 per min.	788 per min.	760 per min.	722 per min.
Air pressure	2in.	2in.	2·8in.	2·6in.	2·35in.
Speed	14·63 knots.	14·42 knots.	13·91 knots.	14·08 knots.	14·26 knots.

Vacuum averaged about 20in., and temperature of stoke-hold 10° above that of open air. (Duration of trial, three hours consecutively.)

A second three hours' run was made a few days later, the speed being 14·08 knots. No indicator diagrams were taken, but the power required to drive the vessel at the mean speed of 14·17 knots had been calculated from data obtained on a number of runs made at progressive speeds up to 12 knots per hour, and the result shows that 169·47 I.H.P. was exerted when steaming at the mean trial speed of 14·17 knots.

The weights of machinery in these boats are worthy of notice.

	lb.
Weight of engine, with pumps, line shafting, stern bearing condenser (outside pipe), propeller, fan, and engine, &c.	2330
Weight of boiler complete, with chimney, fire-bars, &c., and water contained......	3892
Total weight of machinery	6222
or 2·77 tons.	

There was also a water tank, containing 720lb. of water, and, including this under the heading of machinery, the total would be brought up to 7020lb. That is 41·421b. weight of total machinery per I.H.P. when exerting 169·47 I.H.P., and steaming 14·17 knots per hour.

From the particulars here collected the yachtsman will see that in forced draught he has a powerful means provided for gaining speed in his craft; but the penalties that have to be paid for that speed will also be duly noted by the careful owner. But in cases where there is a difficulty in getting steam enough for ordinary running, either through bad ventilation of stoke-hold, insufficient height of chimney, too small a proportion of grate surface, or causes of this nature, a fan would be a valuable addition to the vessel. For short spurts, where quantity of coal burnt is a small consideration, an additional amount of steam can always be obtained with a fan; but it will of course have to be previously considered whether the boiler will bear the higher pressure due to the more rapid evaporation, and the stresses incidental to higher furnace temperature.

In designing new machinery a considerable increase of power for a

given weight of boiler may be obtained by forced draught, but it will be at the expense of the fuel economy and durability of the boiler. If it be determined to maintain the standard of economy, there may still be a gain. It will be due principally, first, to smaller quantity of fire bars; secondly, to the great effectiveness of the heating surface near the furnace, caused by the higher temperature of the gases; and, thirdly, to the more effective mixing of the gases above the furnace, which is a property of forced draught.

Excepting in the first instance, no definite value can be assigned to these different points, it remaining for experiment to give more data from which to draw conclusions.

There is one error in boiler construction that forced draught is likely to encourage, and which the tendency to higher pressures generally accompanying it may perhaps help to exaggerate. Where there is a possible reduction in the grate area, it will be very tempting for designers of boilers to make that reduction by putting in furnaces of smaller diameter; but it must be remembered that it is a very necessary condition of boiler economy that sufficient space should be provided for the mixing of the gases with the oxygen in the air before they are cooled below the temperature of combustion. Mr. James Howden, a Glasgow engineer, has introduced a system of forced draught, with closed-ash pit, in which the air is heated before being forced into the furnace. It has been applied to large vessels with considerable success.

We have already made reference to a method invented by the late Mr. P. W. Willans of applying forced draught in small vessels without closing the stoke-hold. Our illustration (Fig. 125) shows this arrangement.

A casing or chamber, a few inches deep, is fixed to the boiler front over both the usual fire-door and the opening into the ash-pit. The bottom of the casing is below the level of the stoke-hold floor, and is joined to a shoot which brings air from a fan. The front of the casing consists of a large door, which can be turned back or unshipped. When this is in place, the only exit for the air, after it enters the chamber, is through the ash-pit, the firebars, and the funnel. The upper part of the casing is, of course, filled with air under pressure, and at a pressure slightly greater than that inside the furnace. Hence the leak past the fire-door is always in the right direction—inwards. The stoker cannot get at his fire-door for stoking without first removing the outer door, and the moment this is removed the boiler is working by natural draught, for the air from the shoot is as free to pass into the stoke-hold as into the ash-pit, and the pressure in the casing is destroyed without stopping the fan or taking any other precaution. The outer door A, which turns upon a spindle B, and lies upon the floor of the stoke-hold when opened, has a prolongation,

C, which is raised against the top of the air shoot while A is shut, but descends and closes the air shoot as A is opened. The spindle B carries a crank or lever, connected by a slotted link with another lever on the spindle D, which carries the fire door F in such a manner that F does not

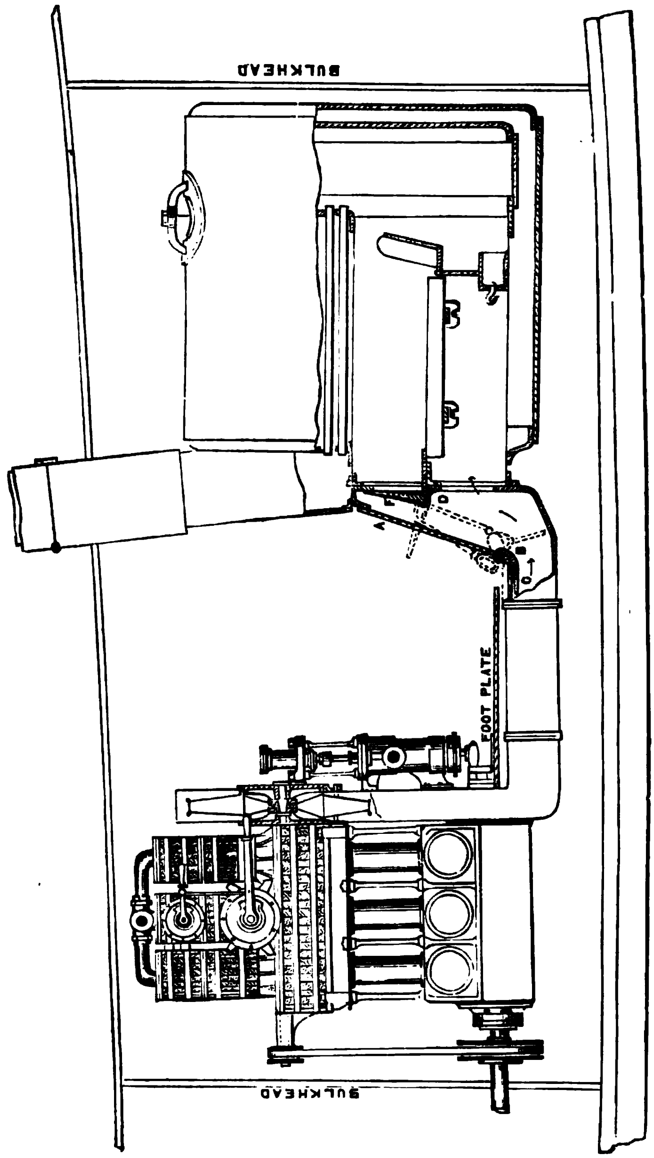


FIG. 125. WILLANS' ARRANGEMENT FOR FORCED DRAUGHT.

commence to open until C has closed the air shoot—though with an easy fit—and has practically shut off the blast.

This arrangement was first applied by Messrs. Willans and Robinson to a small vessel built for Capt. Basil Fisher, and was found to work

very successfully. Since then other applications of the plan have been made.

It should be mentioned that frequently the back of the boiler is next the engine and often bulkheaded off. In such a case there is a separate stoke-hold, and the advantage of this arrangement is that the engine-room is kept free from coal dust and is comparatively cool.

The objection to the plan is that the stoke-hold is usually too small, and consequently ill ventilated. The draught in such a case is less than it would be if the boiler furnaces opened into the larger space represented by the engine room.

Within the last few years the monopoly of steam as a source of power for driving engines on board yachts and other vessels has been attacked by a new description of motor, namely, the oil engine. We have become so accustomed to the marvels wrought by the steam engine during the century now drawing to a close, that we are very apt to forget how imperfect a vehicle water is for transforming the potential energy of coal or other fuel into useful work. The losses incidental to the generation of steam and its use in an engine have been referred to already, and it is not necessary to repeat them. Still, the steam engine remains the only motor used for marine propulsion on a large scale. The oil engine or gas engine —for the oil engine is but a gas engine that generates its own gas—has not many of the sources of loss which are present with the steam engine and boiler. For instance, the great amount of heat which escapes by the boiler chimney represents a loss which is not present in the gas engine. Again, there is no liquefaction of steam in the gas engine cylinder, and the latent heat of evaporation is not an element of waste with the latter type of motor. In addition to this, the temperature of the explosive mixture when ignited is immensely greater than that of steam when entering the cylinder, even at highest pressures, and it is the limited range of temperature possible in an engine which is one of the great reasons of the inefficiency of steam. To set against these drawbacks, there is one great source of loss in the oil engine which is not present with steam. The temperature of the gases when exploded is necessarily raised to so high a degree that it is, practically speaking, impossible to work the marine oil engine unless the cylinder is cooled, and for this purpose a jacket is provided through which water circulates. The most economical method at present known of using gas in an engine is by the well-known Otto cycle. It gives one impulse to every four strokes of a single acting cylinder. Starting with the engine at rest, the first outward stroke would be utilised for drawing into the cylinder gas and air. To produce explosion it is necessary that the mixture should be compressed, and this is effected by the back stroke. Explosion

takes place at the commencement of the next (outward) stroke. The next (inward) stroke—called the scavenger stroke—clears the cylinder of the products of explosion, or so much of them as may be necessary. These four strokes, or two revolutions, complete the Otto cycle. Of course, for three-quarters of the time the work to be done, including the turning of the engine itself, has to be carried on by the energy stored up in the fly wheel, and that accounts for the enormously heavy fly wheels with which gas engines and oil engines are fitted. It will thus be seen that, although steam is a wasteful servant, it has advantages, or rather it has not the disadvantages of other methods of the conversion of heat into work; and, though it is a most fascinating idea to generate the heat, as it is required, inside the cylinder which uses it, the process is attended by difficulties and drawbacks.

In spite of these, however, a number of small vessels have been fitted with oil engines of different types. Some work with the ordinary paraffin or petroleum oils, which can be bought at any oil shop; others require a special oil or spirit of the benzine class. The former have an immense advantage in practical use, as there are restrictions on the transportation of light and highly volatile oils, which make it often difficult to get them in quantity if away from a central source of supply. On the other hand, there is less of the objectionable stench—which, so far as the writer's experience goes, accompanies all oil engines—with those who use the rectified spirit. It is, of course, a great benefit in small boats to get rid of the heavy boiler, with its attendant coal dust; and it is also a nuisance to have to wait a considerable time while steam is raised. The latter disadvantage is largely reduced if a water tube boiler is used with oil fuel and the ordinary form of steam cylinder. Steam engines are more easily reversed and restarted, and generally there is less rattle and vibration than with oil cylinder engines. Whether for small powers the oil engine or the steam engine is preferable depends largely on the predilection of the owner. There is this, however, to be said in favour of the former, that it is comparatively a new departure, and the possibilities of improvement are, doubtless, great; whilst the limitations of the steam engine are pretty well known whether coal or oil be used as the fuel. The purchaser of an oil cylinder engine may therefore console himself for any unpleasant features attendant on the use of these motors by the thought that he is assisting towards the development of an invention that may some day prove a considerable step in the advancement of engineering practice; but, as far as present experience goes in yacht practice, the "oil fuel" and steam generator are in more favour than the vaporising of oil in the cylinders as a motor.

CHAPTER XII.

PROPULSION BY STEAM.

In the preceding chapter the boiler and engine have been dealt with, and now remain to be shown the methods of utilising steam machinery as a means of propulsion. In the early days of steam yachting paddle wheels were used as propellers, but they are now entirely superseded by the screw, and there is so little prospect of the paddle wheel ever coming into use again for yachts that we shall confine our remarks entirely to the screw propeller.

The action of the screw propeller is pretty much the same as that of a screw when turned into a piece of wood, or into a nut; for each revolution of the screw it advances a distance into the wood equal to the distance in a line parallel to the axle D E, Fig. 126, made by one

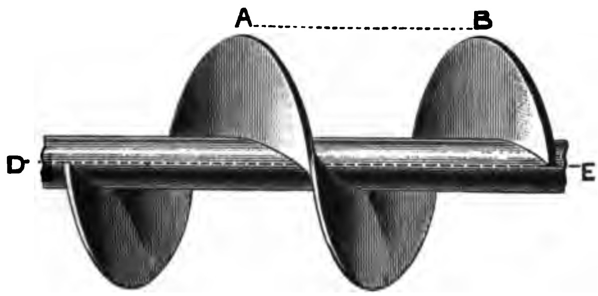


Fig. 126.

complete convolution of the thread. This is termed the pitch, and it will be better understood by reference to Fig. 126, which is an exact representation of Sir Francis Pettit Smith's first screw propeller. In this screw the distance from A to B represents the pitch, and it is clear that one revolution of the screw would advance it exactly the distance A to B, if it were turned in a non-yielding medium.

Pitch is termed coarse or fine according as the distance A B is relatively great or small. The closer the threads of an ordinary carpenter's screw are together the smaller will be its advance into the wood for each revolution, and similarly for the screw propeller in water. Fig. 126 represents a one-threaded screw, but a screw may have two or more

threads. If a string be wound round a cylinder it would form one thread; if another string were wound round between the convolutions of the other this would be two threads, and if another were wound round in between the other two there would be three threads on the cylinder, and so on.

This description answers if, instead of strings, flat ribbands set edgeways, as shown in Fig. 126, were wound round the cylinder or shaft D E. If now we cut away a portion of one of the threads, and leave only a small piece equal, say, to a quarter turn round the cylinder, that would be termed a blade; if we similarly cut away another thread that would be two blades,

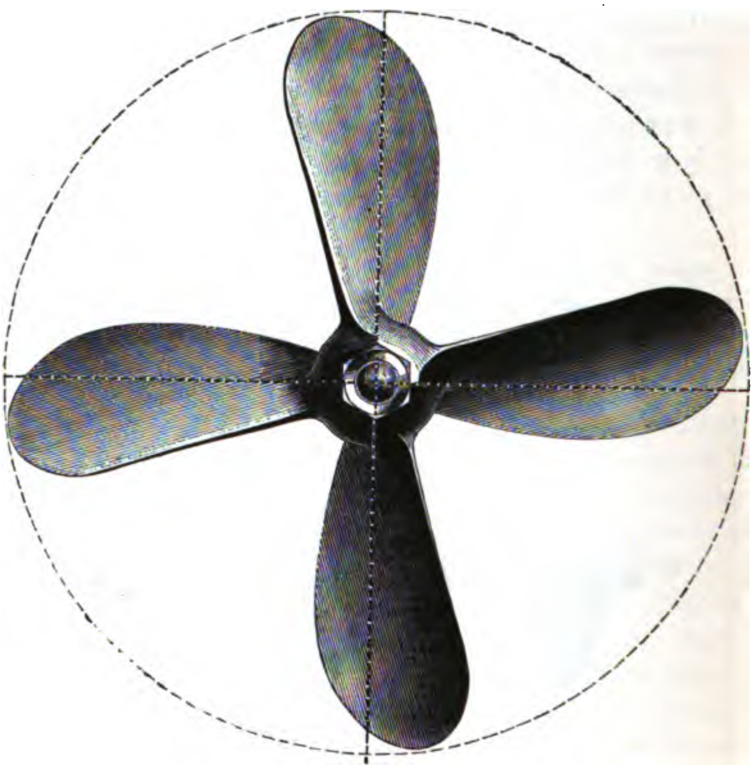


FIG. 127.

and so on. Therefore the number of blades a propeller has indicates the number of threads there would be round the cylinder if the blades made complete convolutions. Fig. 127 represents a four-bladed screw.

The diameter of a screw is the distance from tip to tip of the blades, or the diameter of the circle they describe when revolving. The area of this circle is termed the area of the screw's disc. The area of the blades is the area of the actual propelling surface of the blades.

The length of the screw is the distance occupied by the blades on the shaft D E, Fig. 126.

To measure the pitch of a propeller from the propeller itself is not a difficult operation. The propeller is put on the floor, B, with its driving face downwards. The floor must be perfectly flat and level. Let A, Fig. 128, represent a portion of the blade of a propeller cut off to show the section, say, at 1ft. 6in. from the centre line of the hub or shaft, making the full diameter at that section 3ft. Take a piece of board, D, and shape it to fit the angle of the blade at the diameter named, 3ft. or 1ft. 6in. from the centre line of the shaft. A portion of the board may go beyond the under edge of the blade, as shown at *a*, but whether it does or not will make no difference to the angle. Having obtained the correct angle, take the board and place it flat on the mould loft floor. With a chalk line or straight edge produce the line *a d b*, equal to

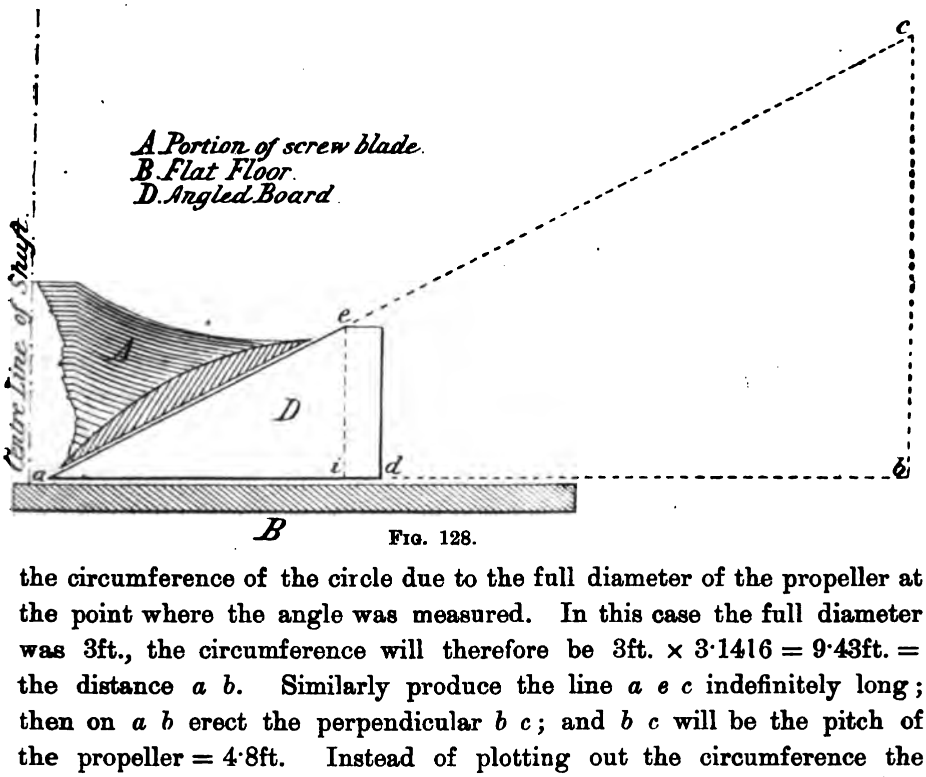


FIG. 128.

the circumference of the circle due to the full diameter of the propeller at the point where the angle was measured. In this case the full diameter was 3ft., the circumference will therefore be 3ft. × 3·1416 = 9·43ft. = the distance *a b*. Similarly produce the line *a e c* indefinitely long; then on *a b* erect the perpendicular *b c*; and *b c* will be the pitch of the propeller = 4·8ft. Instead of plotting out the circumference the angle *a e i* of the board can be measured thus: Accurately measure the base, *a i*, and the perpendicular, *i e*; divide *i e* by *a e*, and the quotient will be the tangent of the required angle, $\frac{i\,e}{i\,a} = \theta$ at *a*.* In this case we have $a\,i = 2{\cdot}8$ft. and $i\,e = 1{\cdot}4252$ft., then $\frac{1{\cdot}4252}{2{\cdot}8} = 0{\cdot}509$, which is the tangent of an angle of 27°, as can be found from a table of tangents.†

* θ is the Greek letter theta, used to denote an angle.

† Law's Mathematical Tables. Weale's Series. Crosby Lockwood & Co. Price 2s. 6d.

The pitch will be the circumference multiplied by the tangent of the angle at a; = 3ft. diameter × 3·1416 × ·509 = 4·8ft. To facilitate the foregoing calculation, the annexed table has been computed from the formula = diameter × 3·1416 × tangent, the diameter being taken as 1ft. If, therefore, the angle of the blade be found at any diameter, and that diameter be multiplied by the number found in the table corresponding to the angle, the required pitch will be the result. For instance, if the angle be 27° (as just referred to), and the diameter 3ft., take from the table opposite 27° the numbers 1·601, and 1·601 × 3ft. = 4·8ft. = the pitch. (It will be noted that the table has been calculated for $\frac{1}{4}$, $\frac{1}{2}$, and $\frac{3}{4}$ degrees, as shown by 15′, 30′, 45′.)

A rough-and-ready way of measuring an angle is with an ordinary carpenter's rule with 1ft. arms. Take the rule, and place it at the diameter of the blade of which the angle is required, as shown in Fig. 129. (It

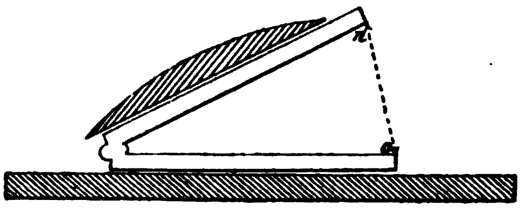


FIG. 129.

does not matter whether the joint of the rule extends beyond the edge of the blade or not, the angle will be correctly taken by the rule all the same.) Measure the opening of the rule at the extreme points of the legs on the inside at $o\ n$. The angle can be found from the following table:—

ANGLES FOUND BY OPENING A 2FT. JOINTED RULE.

Opening of rule in inches (inside).	0		⅛		¼		⅜		½		⅝		¾		⅞	
	°	′	°	′	°	′	°	′	°	′	°	′	°	′	°	′
00	00	00	0	36	1	12	1	47	2	23	2	59	3	35	4	11
1	4	46	5	22	5	59	6	34	7	10	7	46	8	22	8	58
2	9	34	10	10	10	46	11	22	11	58	12	34	13	10	13	46
3	14	22	14	58	15	34	16	10	16	46	17	22	17	59	18	35
4	19	12	19	48	20	24	21	0	21	37	22	13	22	50	23	27
5	24	3	24	39	25	16	25	53	26	30	27	7	27	44	28	21
6	28	58	29	35	30	12	30	49	31	26	32	3	32	40	33	17
7	33	54	34	33	35	8	35	46	36	25	37	3	37	40	38	18
8	38	56	39	34	40	12	40	50	41	29	42	7	42	46	43	24
9	44	4	44	42	45	21	45	59	46	38	47	17	47	56	48	35
10	49	15	49	54	50	34	51	13	51	53	52	33	53	13	53	53
11	54	34	55	14	55	55	56	35	57	16	57	57	58	38	59	19
12	60	0	60	41	61	23	62	5	62	47	63	28	64	10	64	52
13	65	35	66	18	67	1	67	44	68	28	69	12	69	55	70	38

The pitch of each blade of the propeller should be found at several diameters, including the diameter at the tip of the blade. The mean pitch of all the blades will be the pitch of the propeller as used in the calculations of speed. In propellers of 3ft. to 6ft. diameter the pitch should be found at 3 diameters; 8ft., 4 diameters; and so on.

If the propeller is fitted to the vessel on the shaft, the angle of the blade can be found in the manner described, by fixing a straight edge at right angles to the shaft to represent *B* (Fig. 128).

TABLE FOR FINDING PITCH OF A SCREW PROPELLER.

Angle of blade. Deg.		15′	30′	45
1	·055	·068	·081	·096
2	·110	·122	·135	·150
3	·164	·177	·190	·205
4	·220	·234	·248	·262
5	·275	·290	·302	·316
6	·330	·344	·358	·372
7	·386	·400	·414	·430
8	·440	·454	·468	·482
9	·498	·512	·526	·540
10	·554	·568	·582	·596
11	·611	·625	·640	·654
12	·668	·682	·696	·710
13	·726	·740	·755	·770
14	·783	·798	·813	.827
15	·842	·856	·871	·886
16	·901	·916	·931	·946
17	·961	·976	·991	1·006
18	1·021	1·036	1·051	1·066
19	1·082	1·097	1·113	1·128
20	1·143	1·159	1·175	1·190
21	1·206	1·222	1·238	1·253
22	1·269	1·285	1·301	1·317
23	1·334	1·350	1·366	1·382
24	1·399	1·415	1·432	1·448
25	1·465	1·482	1·498	1·515
26	1·532	1·549	1·566	1·584
27	1.601	1·618	1·635	1·653
28	1·670	1·688	1·706	1·724
29	1·741	1·760	1·777	1·796
30	1·814	1·832	1·851	1·870
31	1·888	1·905	1·923	1·944
32	1·963	1·981	2·000	2·019
33	2·038	2·057	2·077	2·098
34	2·120	2·140	2·161	2·181
35	2·200	2·220	2·241	2·261
36	2·282	2·302	2·324	2·345
37	2·367	2·388	2·410	2·432
38	2·456	2·476	2·498	2·521
39	2·544	2·565	2·588	2·613
40	2·637	2·660	2·684	2·707
41	2·731	2·755	2·778	2·803
42	2·828	2·852	2·877	2·903
43	2·930	2·955	2·981	3·008

TABLE FOR FINDING PITCH OF A SCREW PROPELLOR—*continued.*

Angle of blade. Deg.		15′	30′	45′
44	3·034	3·060	3·087	3·115
45	3·142	3·169	3·197	3·225
46	3·253	3·281	3·310	3·338
47	3·365	3·394	3·424	3·452
48	3·487	3·515	3·546	3·578
49	3·614	3·644	3·677	8·710
50	3·744	3·776	3·810	3·844
51	3·878	3·911	3·946	3·982
52	4·020	4·055	4·093	4·130
53	4·169	4·207	4·245	4·285
54	4·324	4·364	4·404	4·446
55	4·487	4·527	4·568	4·610
56	4·655	4·698	4·743	4·790
57	4·837	4·883	4·930	4·980
58	5·027	5·075	5·125	5·178
59	5.228	5·280	5·333	5·388
60	5·441	5·496	5·552	5·610
61	5·667	5·725	5·786	5·848
62	5·908	5·971	6·035	6·100
63	6·166	6·233	6·301	6·370
64	6·440	6·510	6·583	6·660
65	6·737	6·817	6·896	6·976
66	7·057	7·137	7·222	7·312
67	7·401	7·492	7·586	7·680
68	7·776	7·876	7·980	8·080
69	8·183	8·285	8 395	8·514
70	8·631	8·746	8·860	8·990

To measure the area of the surface of the blades, proceed thus: Divide the blades into an equal number of intervals as shown in Fig. 130. The lines, or ordinates, are drawn parallel to the hub or centre line of shaft C, and perpendicular to a straight edge, A B, placed at right angles to the centre line of the shaft C. The breadths of the blade at the several points 1, 2, 3, 4, 5, 6, 7, are then measured, and the area determined by

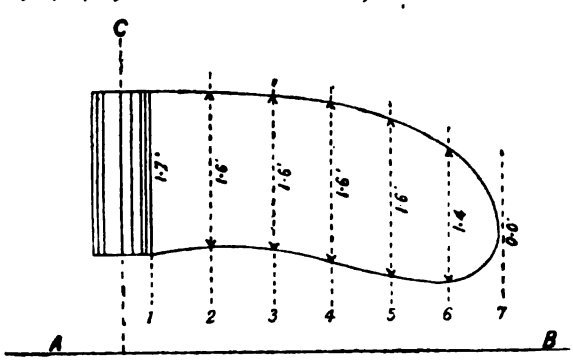


FIG. 130.

"Simpson rule," as described further on. In this case (Fig. 130), if the ordinates were of the lengths given and the interval between the ordinates ·6ft., the area of the surface of the blade would be 5·137 sq. ft.

The speed of the screw is the theoretical distance it advances in a line with its shaft (D E, Fig. 126) for a given number of turns in a given time. Say the pitch is 7ft., and the number of revolutions 200 per minute, then $200 \times 7 = 1400$ft.; that is, the screw would have advanced 1400ft. in one minute. This would be reduced to knots per hour as follows, taking the Admiralty standard knot of 6080ft.:

$$\text{Speed of screw} = \frac{200 \times 7}{6080} \times 60 = \frac{200 \times 7}{101{\cdot}33} = 13{\cdot}8 \text{ knots.}$$

A rule in common use for making hasty calculations is

$$\text{Speed of screw in knots} = \frac{\text{Pitch} \times \text{revolutions}}{100}$$

Thus for 7ft. pitch, and 200 revolutions per minute, we should have $\frac{7 \times 200}{100} = 14$ knots per hour.

The slip of a propeller is the difference between its speed and the speed of the ship it propels. If the screw were turned into wood or into a nut there would be no slip; but as water is a yielding medium, and, moreover, as a screw revolving close under the stern of a vessel is subject to a variety of influences, there must always be slip.

If, as just shown, the speed of the screw is 13·8 knots per hour, and the speed of the ship only 11 knots, the difference would be 2·8 knots per hour, and this would be the slip. The slip is usually expressed as a percentage of the speed of the screw. Let S = speed of the ship, and s speed of the screw, then the slip per cent. k will be $k = \frac{s - S}{s} \times 100$. Or, taking the quantities previously used:

$$\frac{13{\cdot}8 - 11}{13{\cdot}8} \times 100 = 20{\cdot}3 \text{ per cent.};$$

or expressed as a fraction of the speed:

$$\frac{13{\cdot}8 - 11}{13{\cdot}8} = {\cdot}203.$$

For a given number of revolutions (R), pitch (P), and slip per cent. (k) of the propeller, the speed of the ship (S) will be computed from

$$\text{Speed of ship} = \left(\frac{R \times P}{101{\cdot}33}\right) - \left(\frac{s}{100} \times \frac{R \times P}{101{\cdot}33}\right)$$

The speed of the ship and the slip per cent. (k) being predetermined, the required speed of the screw propeller will be found from

$$\text{Speed of propeller} = \frac{S}{100 - k} \times 100.$$

The revolutions required for a given speed of the ship, and speed and pitch of the screw propeller, will be ascertained from

$$\text{Revolutions} = \frac{\text{Speed of Screw}}{P} \times 101{\cdot}33.$$

Or if the slip per cent. of the propeller is only known

$$\text{Revolutions} = \left(\frac{10133 \times S}{P \times 100 - k}\right)$$

The pitch required for a given speed of the ship, and revolutions, and slip per cent. of the propeller would be found from

$$\text{Pitch} = \left(\frac{10138 \times S}{R \times 100 - k}\right)$$

The slip to which reference has been made is termed the *apparent* slip. The real slip cannot in practice be accurately calculated. The real slip is equal to the sternward motion imparted to the water, owing to its yielding nature, by the propeller; but, inasmuch as a vessel when she moves ahead carries a forward wake with her, the propeller is always rotating in water previously set in motion in the direction of the motion of the vessel. The speed of the following motion of the water or speed of the wake, as it is usually called, is dependent on the form of the vessel, and, although there have been rules formulated for determining this speed, they do not appear to be reliable. For all practical purposes it is sufficient to know that the finer the run the smaller will be the velocity of the following current. In reference to this feature in screw propulsion, the late Professor Rankine thus summarised the effect of a propeller working in disturbed water: "The change of pressure produced in the water by the action of the propeller on it, is transmitted to some part of the ship's bottom, and thus the resistance of the ship is altered. The alteration of resistance so produced constitutes a difference between the *total thrust* and the *effective thrust* of the propeller. The effect is always to produce a waste of power when the propeller works in water previously set in motion by the vessel; in other words, when there is a difference between the real and apparent slip. When the propeller works in water set in motion by the ship there is, in the first place, a loss of work proportional to the real slip of the propeller relatively to that moving water, and then a further loss of work proportional to the square of the previous velocity of the water."

It so happens occasionally that there is apparent negative slip, that is to say that the ship appears to have a greater speed than the speed of the propeller; as, for instance, she might have, if assisted by sails or when steaming before a strong wind, or from being assisted by extra propellers in the form of paddle wheels, and in such cases it means that the vessel is towing or dragging the screw through the water. If the propeller always rotated in motionless water, negative slip could only exist under some such conditions as just stated; but, whatever the cause of its appearance, it is

difficult to understand that its presence does not involve the phenomenon of the ship dragging her propeller.

When negative slip was first noticed it was concluded that there had been some errors in measuring the pitch angles, or that the pitch altered during revolution owing to the flexibility of the blades. These, no doubt, are the true explanations of negative slip in most cases, but its existence is also, to some extent, involved in the fact that there is always a following wake or current of greater or less velocity at the stern of a ship, and also that there may be an increase of thrust in consequence of the difference in pressure before and behind the propeller due to the suction of its rotating race. But whatever conditions are involved in the alleged possible negative slip we have never yet come across an instance of it in a yacht.

The theory of the following current is that additional thrust is obtained from it; the theory is surrounded by many complications, but practically it amounts to this: The following current is intercepted by the screw which receives the pressure on its blades which otherwise would have been expended on the stern or quarters of the ship. The first effect is to increase the resistance of the ship by the withdrawal of the pressure on the quarters; but the loss in this respect might mostly be made up by the pressure on the propeller. This pressure on the propeller would have the effect of reducing the number of times the engine could turn it round—in other words, the speed of the propeller would be reduced, but its thrust would be augmented. In such a case negative slip might or might not appear according to whether the reduction in the speed of the propeller exceeded or not the reduction in the speed of the ship due to the abstraction of water pressure around her stern. This much, however, seems certain, that the net result of negative slip or exceedingly small slip is a falling off in the speed of the ship from what it ought to be for any given horse power.

It would seem, from the experiments which have been made, that negative slip is most likely to occur when the blade area is large and when the pitch is small relative to the diameter, because such propellers have small real slip, and because they are at a more favourable angle for facing the pressure of the following wake; also, if the propeller is placed close under the quarters, it largely adds to the causes which produce negative slip, as will be gathered from what has been said. It, however, may be an advantage for the propeller to work in a following wake, providing it is not placed so close to the quarters as to interfere with the water pressures on the quarters, and if the form of the after body is adapted for the purpose. That is to say, if by working in a

following wake the same speed could be obtained for a smaller number of revolutions, there would be a gain in engine economy; but this would not account for negative slip. The resistance of the ship must always balance the actual force employed in propelling her; but true negative slip would involve the absurdity that a ship can be propelled at a given speed by a smaller force than her resistance at that given speed.

With regard to the smallness of slip depending on a large diameter and relatively small pitch, it would appear that this need not necessarily mean that such a propeller would be more efficient than one of smaller diameter and greater pitch, or of similar diameter but of greater pitch. The percentage of apparent slip is therefore not a safe test of the efficiency of propellers, and, so far as our present knowledge goes, the only reliable test is the result of any particular propeller on the speed of a ship with any given engine power.

A great variety of shapes and forms has been given to screw blades, and it would seem that they can be of almost any shape the fancy might dictate. One feature, however, seems pretty certain, that the inner portion of the blade next the shaft or boss is almost useless, as the friction and resistance derived from it is greater than the thrust it gives. It is, therefore, found an advantage to have a large round boss, as introduced by Mr. Griffiths.

In yachts built of wood, with very thick stern posts and rudder posts, the aperture for the screw should be somewhat larger than it is in an iron or steel yacht, to admit of the propeller being kept some distance from the rudder post, so that the eddies around it may be avoided; or, in other words, so that there is an uninterrupted flow of water to the propeller, so that the water acted upon may be undisturbed by eddies. The aft side of the stern post and the rudder post should be bevelled away as much as possible, or rounded away so as to form an oval section, in order to reduce the eddy making.

This question of the diameter of boss and the small influence of root of blade is a very open one. The steam yacht Fauvette, 420 tons, designed by the author, had four shiftable blades which involved a very large boss. The speed fell short of what was expected by nearly a knot per hour. The screw was discarded, and one fitted with fixed blades and cast in Stone's patent metal, the pattern of the propeller being supplied by Messrs. W. White and Son, of Cowes. The length of the blades was increased one foot by the decrease in the diameter of the boss, but the pitch remained practically the same, the most marked difference being that the propeller weighed one ton less than the original propeller, and

increased the speed of the yacht from 14·6 knots to 15·6 knots per hour. (See Fauvette among the plates of steam yachts). This is not a new experience, but can be taken as an addition to the evidence that the diameter of boss and length of blade largely influence the efficiency of screw propellers.

The form of the propeller fitted to Fauvette by Messrs. W. White and Sons, of Cowes, and which gave such excellent results, was somewhat of the form shown on Plate IX.

The dimensions of Fauvette are as follows :

Length over all	189ft. 9in.
Length on L.W.L.	160ft.
Breadth	22ft. 8in.
Depth, top of beams to top of floors	15ft.
Draught of water aft	12ft. 11in.
Area of L.W.L.	2536 sq. ft.
Area of mid-section	181 sq. ft.
Displacement on trial, bunkers and tanks full, and boats in davits	491 tons.
Screw propeller, diameter (White's)	10ft. 9½in.
„ „ pitch mean	17ft. 8in.
Surface of all the blades (four)	45 sq. ft.
Revolutions on trial	100
Indicated horse power	1150
Speed (mean) in knots	15·6
Slip per cent. of propeller	10·6
Total weight of machinery, steam up	167 tons.

With regard to area of blade surface, it would seem that it has been the practice to give too much area and too little pitch, a matter to which the late Mr. Froude was the first to draw attention. The rule formerly given in some of the text books for apportioning blade area for maximum speeds, is as follows :

$$C \times \sqrt{\frac{\text{I.H.P.}}{\text{Revolutions}}}$$

The maximum I.H.P. and revolutions being used in the expression, C is a constant of the following assigned value:

Four blades, 15.

Three blades, 13.

Two blades, 10.

These constants are, however, much too high in yacht practice at present, although they agree pretty well with the apportionment of blade area in the Royal Navy. So far as propellers for yachts of fine under water body form are concerned, a sufficient blade area is in practice found by the formula with a constant of about 10 for either three blades or four blades, and a two-bladed propeller is seldom found in a yacht which is not termed an auxiliary such as Tern or Lady Nell, as will be gathered from the following table:

Yacht.	No. of blades.	Blade surface in sq. ft.	Value of constant. C.	Yacht.	No. of blades.	Blade surface in sq. ft.	Value of constant. C.
Fair Geraldine	3	16·0	10·8	Primrose	4	6·3	8·4
Aries	4	14·6	9·4	Marchesa	4	17·0	11·3
Capercailzie	4	32·2	13·0	Linotte	4	11·0	10·0
Celia	4	3·9	9·8	Imogen	4	37·0	11·1
Nomad	4	7·8	9·8	Olivia	3	11·0	12·0
Oriental	3	13·5	9·0	Cressida	3	17·2	10·8
Amazon	4	7·5	8·4	Lady Nell	4	36·5	13·6
Malikah	4	23·2	11·8	Gertrude	3	17·2	12·4
Fire Fay	4	23·7	10·2	Paulina	4	24.0	9·9
Eionag-na-Mara	4	25·0	11·6	Samara	4	12·0	9·1
Tern	2	4·0	9·0	Fauvette	4	45·0	13·0
Mera	4	24·0	12·8	Speedy	4	16 0	9·4

Too little blade area is generally indicated by an abnormal amount of slip at high speeds, with small indicated thrust; and too much area is shown by smallness of slip, and in such cases the friction from the propeller would be absorbing an undue amount of the power developed by the engine.

Two, three, or four blades seem to be pretty nearly equally efficient, but the greater the number of blades the coarser should be the pitch; and if the vessel is full aft it would appear to be an advantage if the blades are bent sternwards, as the action of the blades on the water would then have a less injurious effect in withdrawing pressure from the quarters (see farther on).

One great advantage of a number of blades is the effect they have in reducing vibration, and the shocks incident to the revolving of two blades with pressures varying at different parts of the revolutions due to differences of immersion and water thrown by the blades on to the dead wood, and other causes.*

With regard to friction, the blades should be made as smooth as possible, especially the front edges, which should be quite knifelike in sharpness, and the general thickness of the blades should not exceed what the strength requires. There would seem to be some advantage if the blades are elastic, and bend whilst revolving, especially in the case of small vessels; and Messrs. Yarrow have recorded a case within their experience of torpedo boat propulsion where, by submitting a thin elastic blade for a perfectly rigid one, the speed was altered from 17½ knots to 19 knots.

A screw of very large diameter and great blade surface, revolving at a high velocity, would by frictional resistance cause a waste of engine

* That one of the chief sources of vibration is from the shocks incidental to the water being thrown upon the sternpost and dead wood there is no doubt. One of the effects of cutting away the whole of the dead wood aft as practised by Mr. J. S. White in his fast steam launches, has been to almost annihilate vibration. It should also be noted that vibration diminishes largely at the higher speeds attained by torpedo boats.

power greater than a screw of increased pitch and somewhat decreased diameter and blade surface. A remarkable instance of this is recorded by Mr. J. Wright concerning H.M.S. Iris.*

This ship was tried with several propellers, the most marked differences being as follows: (It is worthy of note that the constants obtained by the formula just now given were respectively 21 for No. 1, 16 for No. 2, and only 12·3 for No. 3.)

	1	2	3
I.H.P.	7503	7714	7556
Speed of the ship	16·58	18·57	18·59
Diameter of screw	18ft. 6½in.	16ft. 3½in.	18ft. 1½in.
Pitch of screw	18ft. 2in.	20ft.	21ft. 3in.
Number of blades	4	4	2
Area of the blades	194·4	144	112
Revolutions	91.0	97	93
Slip per cent	1·6 (neg.)	3·0	5·0

The real slip of No. 1 propeller was probably less than either of the other two, and therefore showed a negative slip although operating in the same current or wake at the stern. The No. 1 propeller can therefore be regarded as the most efficient, judged by the slip, but its frictional resistance absorbed so much of the engine power that, judged by its effect on the speed of the ship, it proved the least efficient.

If by working with a coarser pitch the same speed can be obtained for the same number of revolutions, there will obviously be a great saving of wear and tear in the engine, although the load on the shaft per revolution would be greater; and there may be some gain in the efficiency of the engines, as the loss in pressure at the admission ports would be diminished at the slower piston speed.

If any great change is made in the steam pressure in the boiler by diminishing the load on the safety valve as the boiler deteriorates, it may then be necessary to reduce the pitch of the propeller. For any given *weight* of steam the *smaller* the pressure the *greater* the volume, and to utilise the same weight of steam in the same time a greater number of revolutions must be made, and perhaps the steam ports made larger. It will thus be seen that the pitch of the propeller may effect the efficiency of the engine; and, working capriciously at varying pressures, may also vary the efficiency of the engine relative to the quantity of steam used.

The results of some experiments made by Messrs. Yarrow with a torpedo launch were published in 1879,† and show how uncertain

* *Vide* Transactions of the Institution of Naval Architects, 1879.
† *Vide* the *Engineer*, Aug. 29, 1879.

the means of forecasting results from what is known of the action of the propeller must be.

		1	2	3	4	5	6	7	8
		Ft. in.	Ft. in.	Ft. in.	Ft. in.	Ft. in.	Ft. in.	Ft. in.	Ft. in.
Diameter		5 6	5 6	5 2	4 4	4 4	3 11	3 10	3 6
Pitch		4 6	5 0	5 0	5 0	6 0	6 0	5 9	7 0
Area of blade surface in sq. ft.		3·4	3·8	2·7	3·8	2·9	3·5	4·3	2·7
Number of blades		2	3	2	2	2	2	2	2
Speed of vessel	75 I.H.P.	—	12·6	12·5	12·4	12·3	12·2	12·0	11·8
	100 ,,	13·6	13·3	13·2	13·1	13·0	12·9	12·4	12·2
	200 ,,	15·2	15·2	15·5	15·3	15·7	15·0	15·0	14·6
	300 ,,	16·8	18·0	17·2	17·8	18·2	17·6	17·4	16·6
	400 ,,	18·0	19·5	—	19·9	19·8	19·8	19·9	18·0
	500 ,,	19·6	—	—	22·4	—	21·4†	—	—
	550 ,,	20·5	—	—	23·0*	—	—	—	—

* This was with 520 I.H.P. only. † This was with 480 I.H.P.

These trials were made under practically the same conditions, the displacement being 27 tons, and the length of the launch about 80ft. water line; and although the I.H.P. might not with these high speed engines be quite accurately calculated, the work done by the engines would vary regularly all through.

Nos. 1 and 8 afford a marked example of what can be done with widely varying pitch, diameter, and blade area; the efficiency of the propellers being practically the same at the higher speeds.

Nos. 5 and 6 afford proof in a remarkable degree of the correctness of Mr. Froude's statement that "a very much longer pitch than has been commonly adopted is favourable to efficiency; and, instead of its being correct to regard a large amount of slip as a proof of waste of power, the opposite conclusion is the true one;" and then referring to a screw working in the following current, he went on to say "one certain effect of it is to make the slip ratio smaller than it really is, and thus mask the conclusion that a greatly reduced slip is a proof of waste of power."

Particulars as to the number of revolutions and amount of slip of Mr. Yarrow's propellers were not published, but a general feature of the trials was that the screws which had the least variation of slip at different speeds, and those with but little slip at low speeds, were without exception bad at the higher speeds. These are indicated as Nos. 1, 2, 3. It will be also seen that those propellers which were best at the highest speeds did not prove the best at the lowest. This feature was also brought to light by the Medusa trials in 1889; a screw of 13ft. 6in. pitch proved to be better than one of 12ft. 3in. in diameter, and 17ft. 3in. pitch at the highest speed, but not at the lowest.

The results of these experiments lead to the conclusion that a

propeller which is good for one speed may not be for another, but the unfortunate feature is that so far as our present knowledge extends the most suitable propeller for any given speed can only be discovered by experiment.

It is seldom, however, that much trouble is taken with the propeller of a yacht after it has been once fitted, although occasionally we hear of one being altered. In the case of the small steam yacht Celia, designed on fine lines by the author, to attain a speed of 10 knots, with 300 revolutions and 60 I. H. P., failing to attain that speed, there seemed to be reason for supposing that the propeller was unsuitable for the purpose. The actual horse power indicated with 300 revolutions was 58, and the speed 9·7 knots, and 30 per cent. slip. This seemed to be an unusual amount of slip, and appeared to imply that the screw was too small, and the indicator diagrams pointed to the fact that the initial pressure was not maintained to the cut-off point of the stroke, consequently the steam was wire drawn, owing to the piston speed being too fast for the size of the admission ports. Accordingly it was determined to provide for running at fewer revolutions, and the screw was taken off, and another fitted of larger diameter, coarser pitch, and about the same surface. The engines indicated about the same power as before, with 220 revolutions, and the speed attained was 10 knots, showing a slip of 17 per cent. The dimensions, &c., of this yacht are as follows:

Length on water line	57ft.
Beam	10ft.
Draught of water with metal shoe	5ft. 10in.
Displacement	27 tons.

FIRST TRIAL.		SECOND TRIAL.	
I.H.P.	58	I.H.P.	—
Diameter of screw	3ft. 4in.	Diameter of screw	5ft. 2in.
Pitch of screw	4ft. 7in.	Pitch of screw	5ft. 6in.
Number of blades	4	Number of blades	3
Blade surface	3·9 sq. ft.	Blade surface	3·8 sq. ft.
Revolutions	300	Revolutions	220
Speed of yacht	9·7 knots	Speed of yacht	10 knots
Slip per cent.	30	Slip per cent.	17
Speed with 200 revolutions	7 knots	Speed with 190 revolutions	9 knots
Slip per cent.	23	Slip per cent.	14
I.H.P.	28		

The gain here with regard to wear and tear of the engines was very considerable, as the desired speed was attained with about 30 per cent. fewer revolutions; and no doubt the machinery was really more efficient when working at the reduced number of revolutions.

A somewhat similar experience occurred with the steam yacht Olivia, built by Messrs. A. and J. Inglis, of Glasgow. The dimensions, &c., of this yacht are

Length on water line	98ft.
Breadth, extreme	15ft.
Draught of water aft	8.8ft.
Displacement	130 tons.

The steam trials with three different propellers gave the following results:

	Diameter.		Pitch.		Total Blade Area.	Revolutions.	Speed.
	Ft.	in.	Ft.	in.	Sq. ft.		
Propeller No. 1	5	6	7	6	7	155	*
" No. 2	5	6	8	0	9	155	9·9
" No. 3	5	6	8	0	11	150	9·9
" No. 3	5	6	8	0	11	150	10·2

* The speed with this propeller was not accurately taken, but it was barely nine knots, and showed an abnormal amount of slip.

The case of the steam yacht Amazon affords another proof of the uncertainty of any forecast of what the actual effect of the propeller will be on the speed of a vessel with a given number of revolutions. The Amazon was designed by the author to attain a speed on the mile of 10·5 knots. In order to insure a high efficiency of the three-bladed propeller, the diameter was made as great as well could be—6ft., the pitch 7·1ft., and area of blade surface 8·5 sq. ft., revolutions 190, and slip 15 to 20 per cent. Upon the first trial, with a strong easterly wind blowing up the mile in Stoke's Bay, 164 revolutions were obtained, which gave a speed of 10·49 knots; the steam pressure in the boiler being 97lb. On the second trial with a smart beam wind, a boiler pressure of 100lb. was easily kept and with 174 revolutions a speed of 11·1 knots was obtained, although the displacement had been increased by 7 tons by putting in ballast.

The slip was only 9 per cent., or a little more than half what had been anticipated with about 190 revolutions; it was plain that the higher number of revolutions could not be accomplished; but, as the expected speed was more than realised, the result could only be regarded as satisfactory. There was, however, very considerable vibration from the three-bladed propeller. To alleviate this a four-bladed screw was fitted, with slightly increased pitch and 1 sq. ft. less blade surface. It was expected that there would be a slight falling off in the efficiency of the propeller, but on the contrary it proved more efficient, a speed of 10·6 knots being obtained with 93lb. steam pressure and 155 revolutions, the slip being

3 per cent. less. This circumstance pointed to the correctness of the contention, first promulgated by the late Mr. Froude, that it is the anterior edges of the blades which do the greatest amount of work (*vide* "Proceedings of the Institution of Civil Engineers," 1871), and in this instance the additional blade more than made up for the loss of surface.

It should be noted that when the third trial was made the weather was very unfavourable, a dead calm prevailing, so that the full boiler pressure of 100lb. could not be obtained; otherwise the result would probably have exceeded that given on the second trial.

SPEED TRIALS OF THE S.Y. AMAZON.

	First Trial.	Second Trial.	Third Trial.
Length on water-line	83ft.	83ft. 4in.	83ft. 4in.
Beam, extreme	15ft. 6in.	15ft. 6in.	15ft. 6in.
Draught water aft	8ft. 3in.	8ft. 2in.	8ft. 2in.
Draught forward	5ft. 6in.	6ft. 2in.	6ft. 2in.
Displacement	77 tons.	84 tons.	84 tons.
Area of mid-section	60 sq. ft.	64 sq. ft.	64 sq. ft.
Area of wetted surface	1380 sq. ft.	1438 sq. ft.	1448 sq. ft.
Diameter of cylinders	10in. and 20in.		
Stroke ditto	16in.		
Diameter of boiler	8ft.		
Length of ditto	7ft. 6in.		
Diameter of furnaces (2)	2ft. 4in.	Ditto.	Ditto.
Grate area	23 sq. ft.		
Heating surface	509 sq. ft.		
Total weight machinery, steam up	25 tons		
Load on safety valves	100lb.		
Mean pressure on trial	97lb.	100lb.	93lb.
Revolutions	164	174	155
I.H.P.	124	Not taken.	125
Vacuum	24in.	25½in.	26in.
Diameter of propeller	6ft.	6ft.	6ft.
Pitch ditto	7·1ft.	7·1ft.	7·4ft.
Surface of all the blades (3)	8·5 sq. ft.	8·5 sq. ft.	7·5 sq. ft.*
Speed of yacht on trial	10·49 knots.	11.1 knots.	10·6 knots.
Slip per cent. of propeller	8·7	9	6
State of weather	Strong head wind and some sea.	Smart beam wind, smooth sea.	Dead calm.

* This was a four-bladed propeller.

A similar steamer, named Linotte, was subsequently designed by the author to obtain a speed of 11·5 knots on the measured mile. Her dimensions are as follows:

Length on load line	88·25ft.
Beam extreme	14·5ft.
Draught of water	6·3ft.
Area of mid-section	54 sq. ft.
Displacement	84 tons.
Diameter of propeller	5·5ft.
Pitch of propeller	8·75ft.
Surface of all the blades (four)	9·4 sq. ft.
Diameter of second propeller	5·4ft.
Pitch ,, ,,	9·6ft.
Surface ,, ,, (four)	11 sq. ft.

SPEED TRIALS OF THE S.Y. "LINOTTE."

	First Run.	Second Run.	Third Run.	Trial with 2nd Propeller.
Indicated horse power	196	128	71	187
Revolutions....................................	175	157	120	165
Speed	11·5 knots	10·7 knots	9 knots	12·05 knots.
Slip per cent. of propeller.............. ..	23·8	20·7	13	22·9

It must not be concluded that the area of the blade surface can be decreased indefinitely; this is by no means the case, and the inefficiency of a propeller caused by insufficient area of blade can be detected by aid of a curve of indicated thrust, as the thrust will undergo but little augmentation as the speed increases. This matter will be treated of further on. In a trial on the measured mile the fact that a strong wind is blowing in the direction that has to be steamed may make the results of the runs very unreliable, especially if the yacht has large deck houses, bridges, masts, and funnel. The retardation when steaming head to wind will be much greater than the acceleration when before the wind, as if the boilers have nothing but natural draught to urge the fires, there will be a falling off in the steam pressure whilst making the run before the wind. Again, if the wind and tide are in the same direction, the run to windward will be longer, and the retardation greater. For these reasons, a trial should be made, if possible, with a beam wind.

It may happen that an abnormal amount of slip will occur when steaming against a head wind, or towing another vessel. This simply means that the resistance of the wind so much retards the vessel's speed that the screw has to work as it were over and over again in the same water, and is in fact to some extent churning the water and making an inefficient thrust. There may also be an abnormal amount of slip if the vessel is bluff in the bow, and from that cause difficult to propel.

Some general conclusions deduced by the late Mr. Froude concerning the propeller were as follows:*

That at moderate speeds, when the resistance varies from speed to speed as the square of the speed, the slip per cent. should be uniform; but in practice there is usually a slight increase of slip as the number of revolutions increase.

When the resistance varies in greater ratio than the square of the speed, there will be a greater increase of slip.

If two ships have the same resistance at different speeds, the screw area which will overcome the resistance, while maintaining a given slip

* *Vide* a paper in the Transactions of the Institution of Naval Architects, 1878, "On Pitch, Slip, and Propulsive Efficiency."

ratio, will be less in the ratio of the squares of the speed for the ship which has the higher speed.

The number of blades for any given length of blades and total area does not appear to much influence slip.

Twin screws have frequently been tried in yachts of shallow draught when a single propeller of sufficient diameter could not be used, so as to obtain the desired speed and take up the full engine power. A separate engine must of course be provided for each propeller. Beyond the advantage indicated, a vessel with two screws can be easily manœuvred, and if one propeller or engine became disabled, the vessel could be propelled with the remaining one. The chief objection to twin screws is their costliness in fitting and in duplicating the machinery.

We have now to consider the propeller in relation to the vessel it has to propel. It has already been pointed out that the currents which close in around the stern of a vessel as she advances through the water exert a forward pressure, and consequently act in balancing the head resistance. The action of the screw is to drive away some of the following current and lessen the pressure around the stern of the vessel, which is equivalent to increasing the pressure at the bow. The larger the disc area of the propeller and the larger the area of the surface of its blades, and the nearer it is placed to the stern, the greater will be the action of the propeller felt in augmenting the resistance in the manner described. As the propellers are usually placed there seems to be no escape from this augmentation, but it grows less and less as the fineness of the afterbody is increased. Increasing the fineness of the afterbody is virtually placing the screw farther aft, and it has been found a great advantage to remove the screw abaft the rudder in very small yachts; but there would be too much risk attending this practice in large vessels, unless some secure means could be devised for supporting the shaft. From some experiments made by Mr. Froude, he concluded that the advantages of placing the screw farther aft away from the sternpost went on increasing until removed a distance of about one-third the breadth of the vessel away; after that there was no gain, and the result pointed to the conclusion that there was an advantage if the screw worked in the wake, providing, as just explained, it did not interfere with the hydrostatic pressure on the vessel due to the currents closing in around her stern.

Beyond this, Mr. Froude endeavoured to determine more exactly the influence the screw has in augmenting the resistance in the manner described. After he had obtained the exact resistance of each model by towing it from the truck across the tank, he had the models towed again at the same speeds, but with another following truck from which was worked a screw immediately under the stern of the model in the

position it would occupy in a real ship, but not actually attached to it; and needless to say the propellers were in each case of suitable size, and made to travel at speeds suitable to the speed of the models. By this most ingenious arrangement he discovered that the augmented resistance amounted to as much as from 40 to 50 per cent.; that is to say, the towing strain on the dynamometer at any given speed was augmented from 40 to 50 per cent.; or, in other words, that the resistance of the models was increased to that extent. These models were "ship forms," and in finer forms the augment of the resistance from this cause was reduced until it was under 20 per cent. These results show that beyond the great resistance due to a very full afterbody, the action of the screw augments that resistance in a greater ratio in the vessel with a full run than it does in a vessel with a long, fine, and clean run; hence it is of greater importance to have the screw far away from the stern in a vessel with a full run than it is with a fine run as already explained.

The foregoing affords an explanation of why a vessel which is made

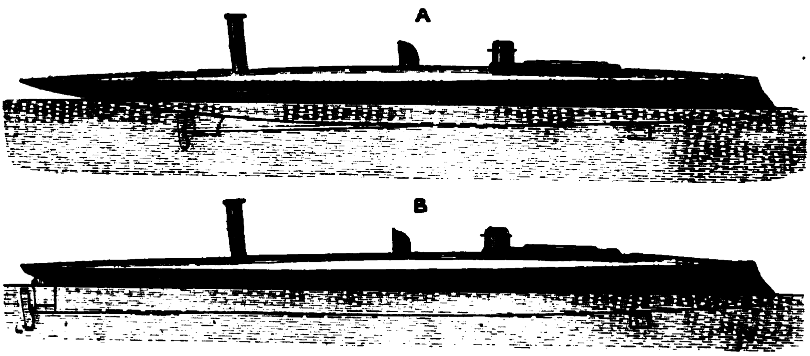


FIG. 131.

full in the run, in order to carry the weight of her machinery aft, is a difficult one to drive.

It has always been considered an advantage to have the propeller deeply immersed, and under some conditions this is correct. At any rate it is a disadvantage for the propeller to work so that the tip of the blades come near to, or above the surface of the water, because the churning action near the surface causes a diminution of pressure or thrust, and, moreover, brings unequal strains on the propeller and shaft due to the unequal pressures. But, beyond this, there is no doubt if the propeller has to work close to the stern it would be an advantage to immerse it so deeply that it did not interfere with the stream line action of the water closing in around the stern. Carrying out this view of the case, Messrs. Yarrow, in 1880, following up an idea of Messrs. Herrischoff, put the propeller under the keel, but some distance forward of the sternpost instead of in the usual

position (see A, Fig. 131). Length of boat 86ft., beam 11ft., displacement 32 tons.

The results fell short of what was expected, and the boat was then tried with the same propeller in the ordinary position (shown by B, Fig. 131). The results obtained were as follows:

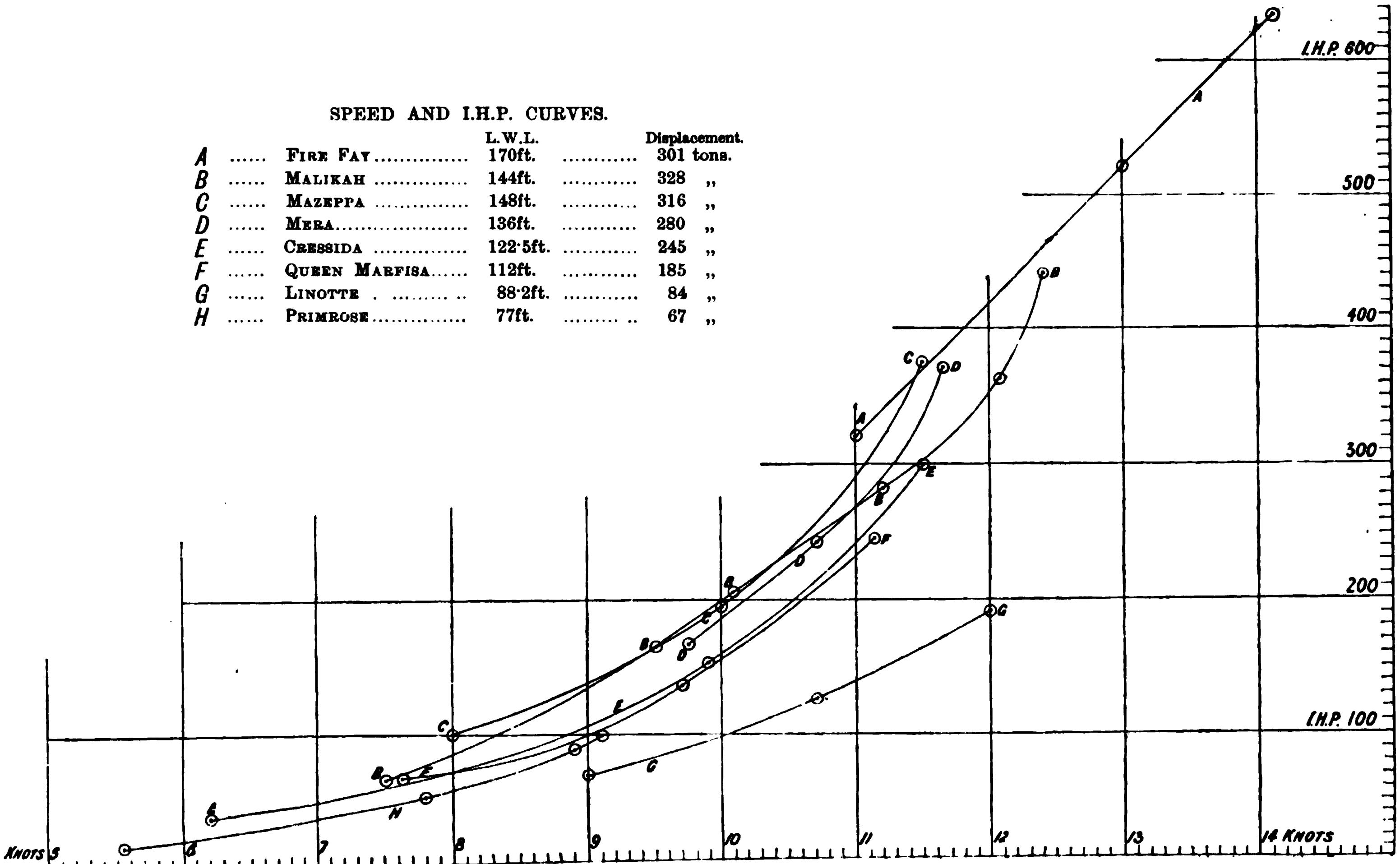
I.H.P.	A Speed.	B Speed.
300	14·8	15·2
400	16·7	18·3
500	18·3	20·7

There is not much doubt that the difference in the speed was mainly attributable to the action of the propeller augmenting the resistance by disturbing the pressure of the closing in currents about the stern. Could the propeller have been carried deeper, a different result might have ensued. With regard to this view of the case several experiments have been tried with jointed shafts, which would admit of the vertical position of the propeller being altered at will, but the advantages do not seem to have been commensurate with the risks run. It is, however, the practice in most fast steam launches to place the propeller so that a portion of its blades comes somewhat below the lowest point of the keel.

From the early days of steam propulsion the co-efficient of performance of steamships has been derived from the expression $C = \frac{V^3 \times D^{\frac{2}{3}}}{\text{I.H.P.}}$; or, M, area of midship section, has been substituted for D, displacement. In this expression it is assumed that the resistance varies as the square of the speed, and that the power required to overcome that resistance varies as the cube of the speed; that the useful power of the steam engine varies as the I.H.P., and that for any given speed the resistance of ships of similar form, but of different dimensions, will vary as the two-third power of their displacements, or as the simple area of their midship sections.

In the chapter on "Resistance" it is made sufficiently clear that even in the case of similar forms the resistance may not vary as the displacement $^{\frac{2}{3}}$, nor as the area of the midship section; and, moreover, it is very wide of the mark to assume that the resistance for different speeds in any given ship will vary as the square of the speeds. How the resistance really does vary has only recently been brought to light by the introduction of "progressive trials," by the aid of which speed curves have been constructed, and the resistance as interpreted by the I.H.P. can be read off for any speed.

So far as steam yachts go the credit is due to Mr. John Inglis, jun. (Messrs. A. and J. Inglis, Glasgow), of introducing these progressive trials with the steam yacht Oriental (see page 297). Since that time there have been numerous other recorded progressive trials, and particulars will be found of them in the annexed diagrams, Figs. 132, 133. The curves on



		L.W.L.	Displacement.
A	Fire Fay	170ft.	301 tons.
B	Malikah	144ft.	328 ,,
C	Mazeppa	148ft.	316 ,,
D	Mera	136ft.	280 ,,
E	Cressida	122·5ft.	245 ,,
F	Queen Marfisa	112ft.	185 ,,
G	Linotte	88·2ft.	84 ,,
H	Primrose	77ft.	67 ,,

Fig. 132.

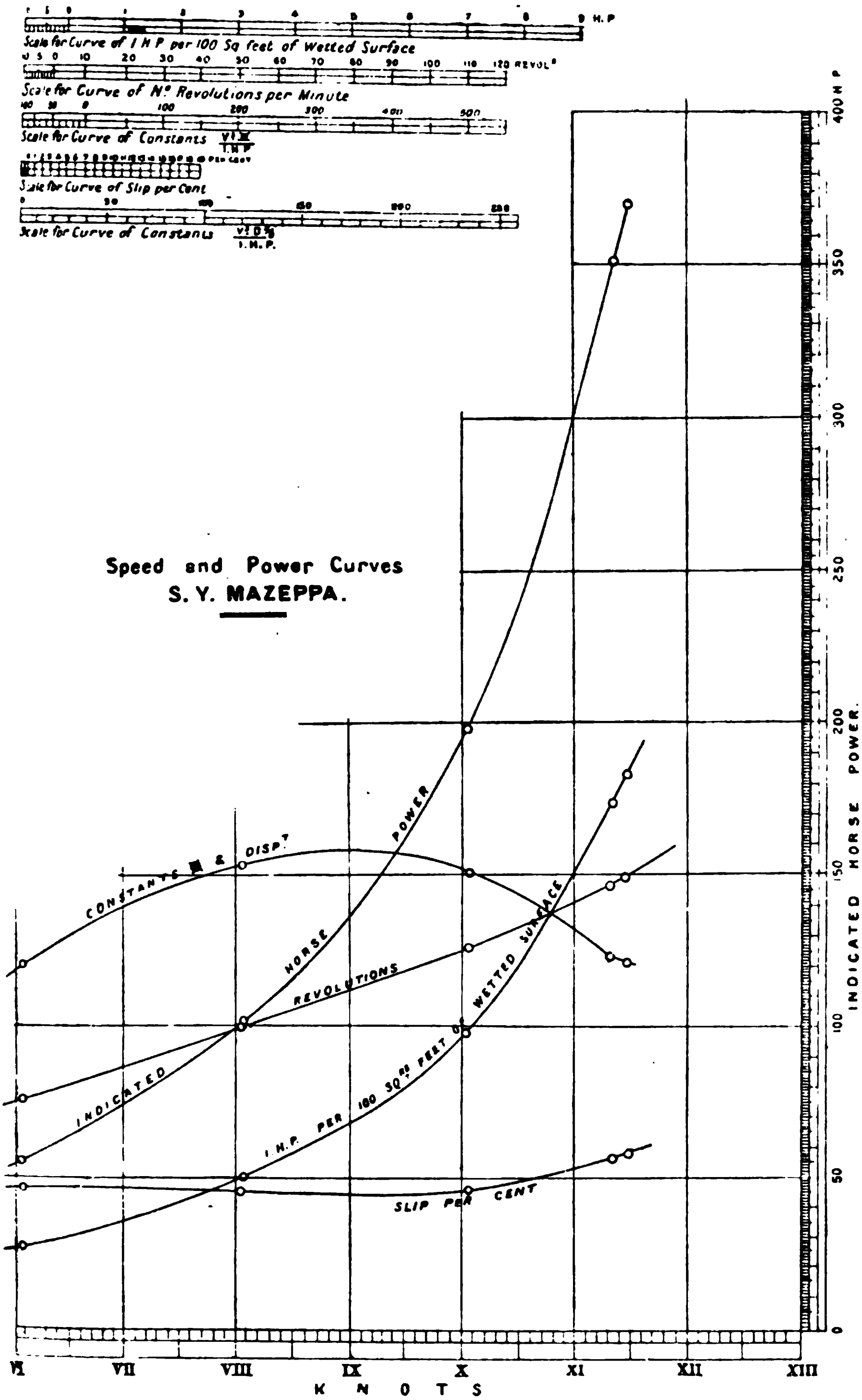

Scale for Curve of I H P per 100 Sq feet of Wetted Surface
Scale for Curve of N° Revolutions per Minute
Scale for Curve of Constants
Scale for Curve of Slip per Cent
Scale for Curve of Constants
Speed and Power Curves
S. Y. MAZEPPA.
CONSTANTS & DISP.T
INDICATED HORSE POWER
REVOLUTIONS
I.H.P. PER 100 SQ.RE FEET OF WETTED SURFACE
SLIP PER CENT
VI
VII
VIII
IX
X
XI
XII
XIII
K N O T S
0
50
100
150
200
250
300
350
400 H P
INDICATED HORSE POWER

FIG. 183.

page 295 will illustrate the main elements of a vessel's performance which came out upon the measured mile and relate to the yacht Mazeppa, designed and built by Messrs. Ramage and Ferguson, of Leith.

The small circles show the speeds on the trial when the I.H.P. was taken. The curves of constants is derived from the expression $C = \frac{V^3 \times D^{\frac{2}{3}}}{\text{I.H.P.}}$. If the law of resistance varied as the square of the speed, the curve of constants would have been a straight line; but it is plain it does not. However, the curve shows when the economical speed was reached by its commencing a downward tendency again after passing 9 knots. Thus, by the aid of a curve of indicated horse power, combined with a curve of constants, we can arrive at the most economical speed with regard to engine power and coal consumption to drive the yacht. In this particular case the variation in the power required to drive the yacht from speed to speed was as follows:

Speed.	Power of speed to which the I.H.P. was proportional.
6	—
8	square.
10	cube.
11·5	4·4 power.

The constants have been more generally used in steamship practice as a guide in designing and putting engines into other vessels, but even with the greatest care in making the comparison between the vessels under consideration the method is very untrustworthy for anything but very moderate speeds; and then a comparison between the wetted surface and the I.H.P. is a better guide.

The co-efficient for Mazeppa would be thus found for her highest speed (and it was unusual to find it for any other).

Her displacement on trial was 316 tons, her speed 11·5 knots, and I.H.P. 375.

$$D^{\frac{2}{3}} = 316^{\frac{2}{3}} = 46{\cdot}4^{*}$$
$$V^3 = 11{\cdot}5^3 = 1521$$
$$\text{I.H.P.} = 375$$
$$\text{Co-efficient} = \frac{1521 \times 46{\cdot}4}{375} = 188.$$

Any other speed (V *a*) for any other given I.H.P. would therefore be found from

$$V\,a = \sqrt[3]{\frac{\text{I.H.P.} \times C}{D^{\frac{2}{3}}}}$$

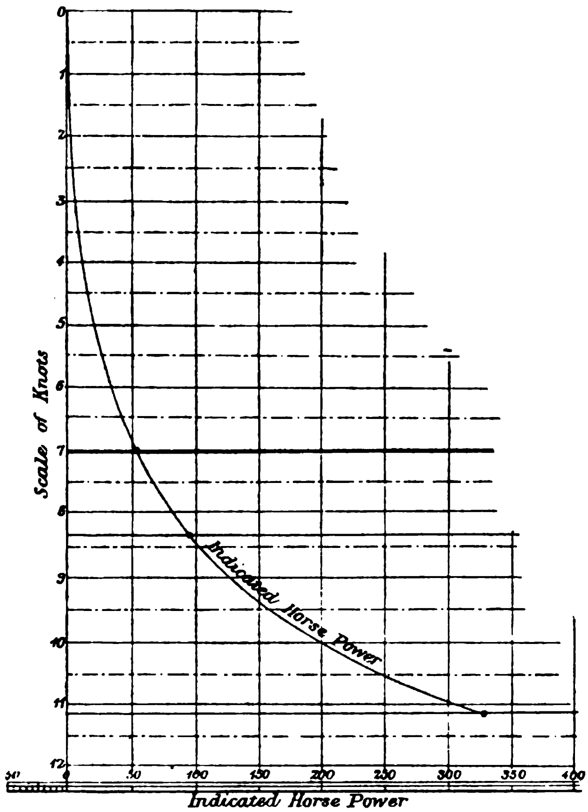
To find the speed of Mazeppa for 100 I.H.P. we should therefore have

$$V\,a = \sqrt[3]{\frac{100 \times 188}{46{\cdot}4}} = 7{\cdot}386 \text{ knots.}$$

* The $\frac{2}{3}$ power of a number is the square of its cube root.

From the trials made with Mazeppa it will be seen that her speed was a little over 8 knots with 100 I.H.P.

Similarly, the speed for any other displacement (the form being similar) would be approximated from the same expression, but as no account would be taken of the speed due to the altered dimensions, the result would probably not accord with practice. One example will suffice to show its unreliability for general use. Fig. 134 shows the speed curve of the yacht Oriental, as constructed by Mr. John Inglis.



Speed Diagram "ORIENTAL."

FIG. 134.

It will be seen that with 306 I.H.P. she made a speed of 11 knots; this would come out in the speed formula, using Mazeppa's co-efficient of 188, as follows, Oriental's displacement being 266 tons:

$$\text{Speed} = \sqrt[3]{\frac{306 \times 188}{266^{\frac{2}{3}}}} = 11{\cdot}128 \text{ knots.}$$

This is a very close approximation, because the co-efficient was derived from a speed when the ratio of its growth was about the same in both vessels. Tried by a lower speed, with 100 I.H.P., and the result is

7·68 knots; whereas Oriental, as will be seen by her curve, made $8\frac{1}{4}$ knots with 100 I.H.P. This discrepancy is due to the fact that the resistance was growing in a much less ratio at $8\frac{1}{4}$ knots in Oriental than it was in Mazeppa at $11\frac{1}{2}$ knots.

Oriental's own co-efficient for 11 knots is 180, and for 11·5 knots this would give an I.H.P. of 350, whereas the direction of her curve shows it should be nearer 375. The difference is due to the fact that at 11 knots the resistance is growing at a higher ratio than the cube of the speed.

The formula is also used for ascertaining the I.H.P. required for any assigned speed and displacement, thus: I.H.P. $= \frac{V^3 \times D^{\frac{2}{3}}}{C}$, or for the same vessel the I.H.P. for different speeds will be proportional to the cubes of the speed.

$$\frac{V a^3 \times \text{I.H.P.}}{V^3} = \text{the required I.H.P. for the different speed } V\,a.$$

But, for reasons already stated, the formula is only to be trusted when the dimensions, displacement, and form are similar, and when there is no great difference in the projected speed and the speed from which the co-efficient was abstracted.

The late Professor Rankine introduced a formula for pre-determining the engine-power required for any given speed based on the area of immersed surface and Sine² of the mean angle of bow entrance, which he termed the augmented surface; but the rule has the fault common to the other; it shows too great a resistance at low velocities, and too small at high, and, moreover, the rule is likely to involve an error, as it is known as a fact that the resistance of oblique surfaces varies nearly as the sine of the angle up to 30° and not as sine².

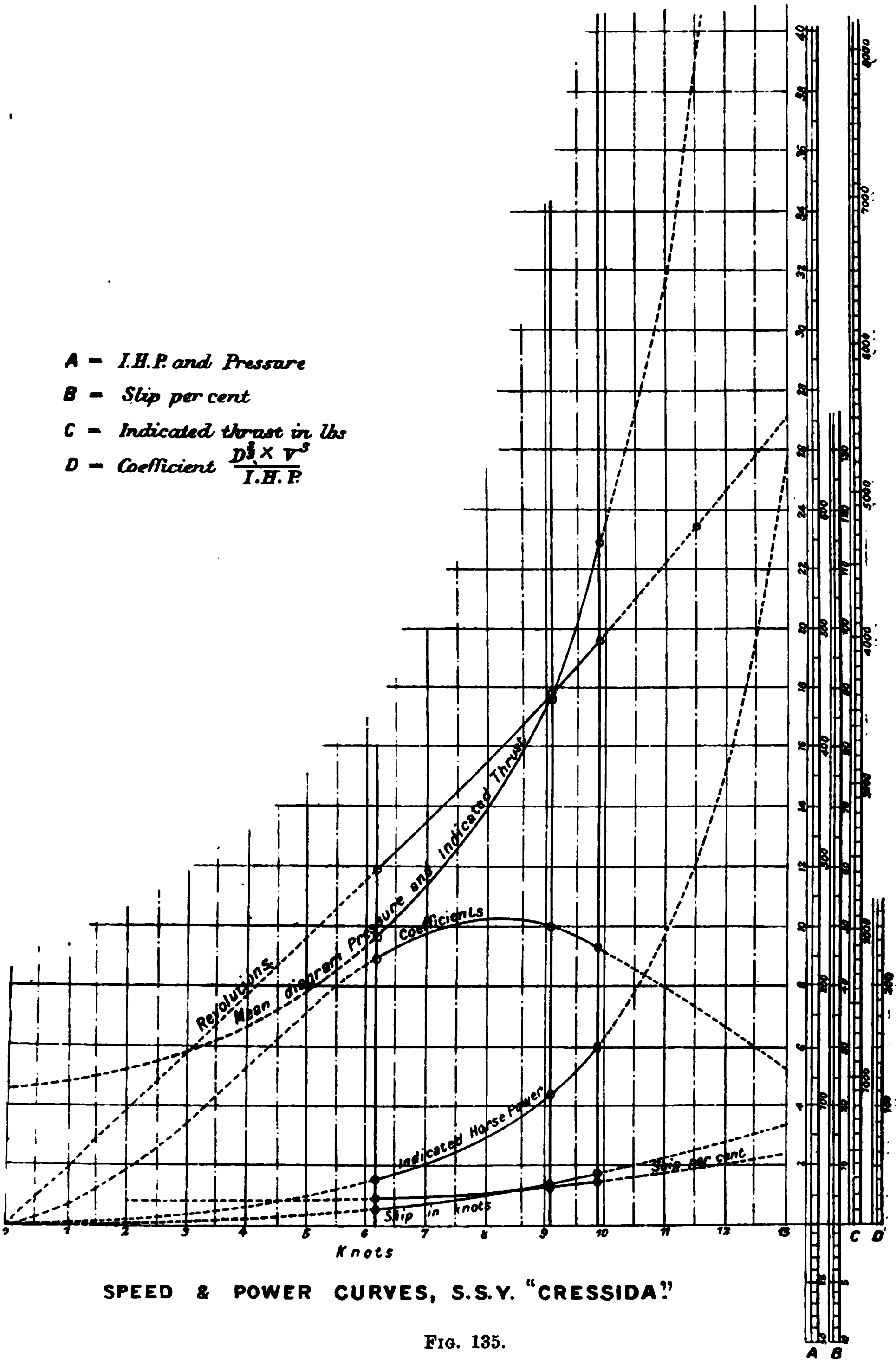
The augmented surface is computed from $1 + (4 \times \sin.\theta^2) + \sin.\theta^4 \times$ mean girth $\times$ length on load line. The I.H.P. that it will take to drive a vessel a certain number of knots per hour is then computed from

$$\text{I.H.P.} = \frac{\text{Augmented surface} \times \text{Speed}^3}{20000}$$

and the speed for any given indicated horse-power will be

$$\frac{\text{I.H.P.} \times 20000}{\text{Augmented surface}} = \text{cube of speed.}$$

20,000 is the "co-efficient of propulsion" for clean painted iron; for clean copper sheathing the co-efficient is greater, being as much as 21,802. The "co-efficient of propulsion" represents the number of square feet of augmented surface that can be driven at one knot per hour by one indicated horse-power, allowing for waste of engine-power. This co-



SPEED & POWER CURVES, S.S.Y. "CRESSIDA."

FIG. 135.

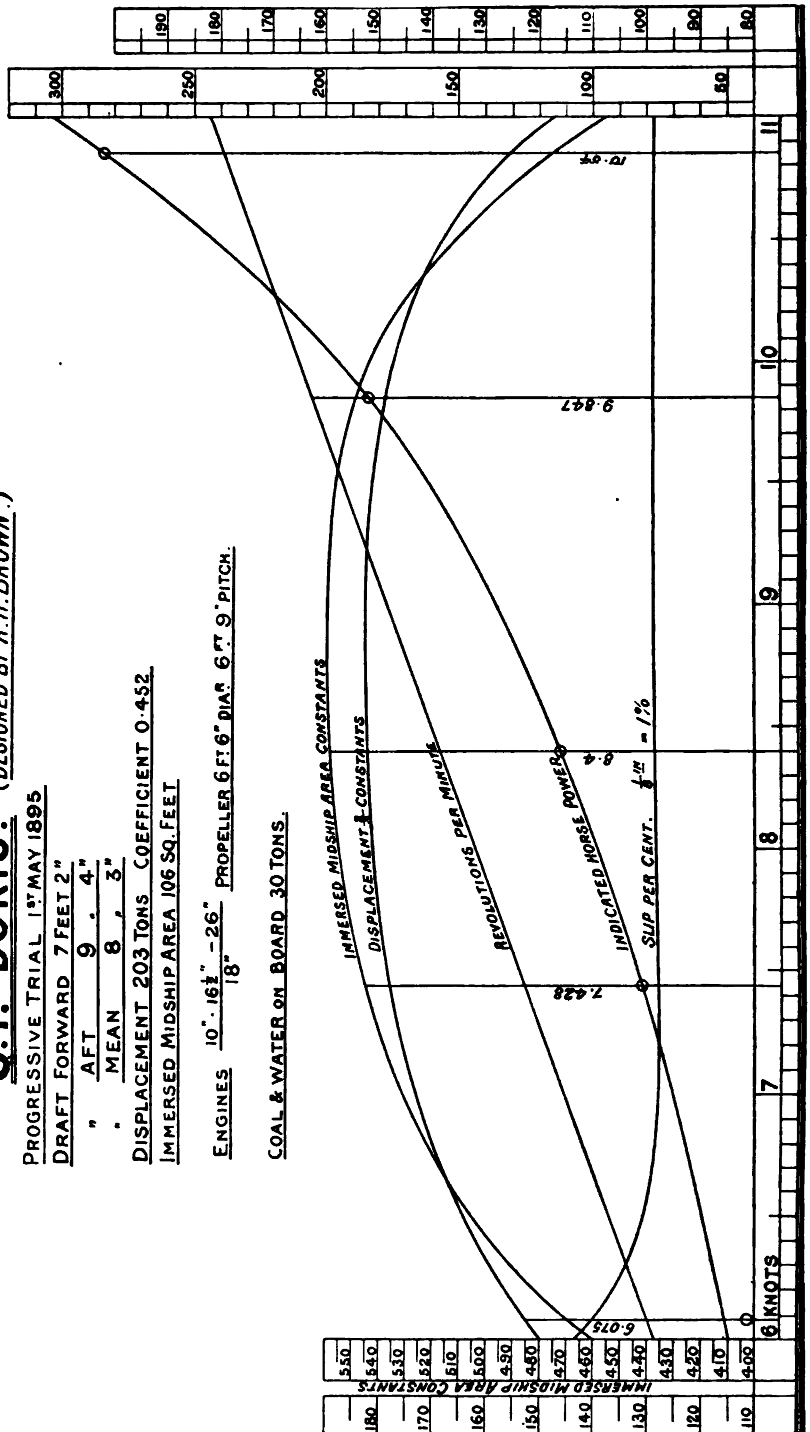

S.Y. "DORIS." (DESIGNED BY A.H. BROWN.)
PROGRESSIVE TRIAL 1ST MAY 1895
DRAFT FORWARD 7 FEET 2"
" AFT 9 " 4"
" MEAN 8 " 3"
DISPLACEMENT 203 TONS COEFFICIENT 0·452
IMMERSED MIDSHIP AREA 106 SQ. FEET
ENGINES 10"·16½"-26" / 18" PROPELLER 6 FT 6" DIAR 6 FT 9" PITCH.
COAL & WATER ON BOARD 30 TONS.
IMMERSED MIDSHIP AREA CONSTANTS
DISPLACEMENT ⅔ CONSTANTS
REVOLUTIONS PER MINUTE
INDICATED HORSE POWER
SLIP PER CENT. ⅛" = 1%
6 KNOTS
7
8
9
10
11
6·075
7·428
8·4
9·847
10·04

FIG. 186.

efficient varies with the relative efficiency of the engines and the conditions of the vessel's bottom.

The probable resistance in pounds is computed from

$$\text{Resistance in lb.} = \frac{\text{Augmented surface} \times \text{Speed}^2}{100 \text{ (for clean paint)}}.$$

The augmented surface rule is thus applied. In the first place, the sines of the angles of greatest obliquity of the water-lines are obtained, and this will be done by drawing tangents from the points of inflection in the water-lines, and measuring the angles made by those tangents with the middle line of the half-breadth plan (see Fig. 137).

All the water-lines will be treated in this way. As only the *sines* of the angles are required, the angles themselves need not be measured, and the sines can be computed thus (Fig. 137): divide $b\ c$ by $a\ b = \frac{b\ c}{a\ b}$ = sine of the angle of greatest obliquity.

The mean girth of the vessel will be obtained by measuring the girths of the various sections on the body plan, as explained farther on.

In the example now given, the calculation applies to the steam yacht Xantha, designed by Mr. John Harvey to attain a speed of 11·5 knots per

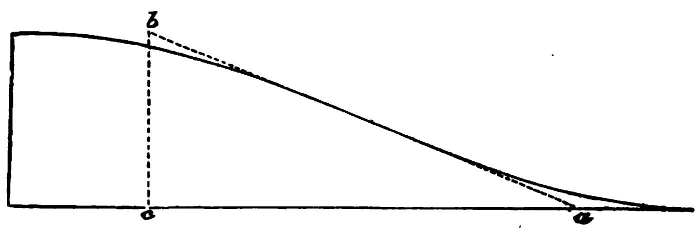


FIG. 137.

hour; her length was 105ft. on the load line; 19ft. beam; 10ft. draught of water; and 180 tons displacement.

CO-EFFICIENT OF AUGMENTATION.

	Angles of greatest obliquity.	Sine of angle of greatest obliquity of water lines	Square of sine.	Fourth power of sine.
L.W.L. 1	15° 22′	·265	·0702	·00492
W.L. 2	14° 37′	·253	·0640	·00409
W.L. 3	13° 25′	·232	·0538	·00290
W.L. 4	9° 51′	·171	·0292	·00085
W.L. 5	4° 18′	·075	·0056	·00003
Keel 6	0° 00′	·000	·0000	·00000
			6)·2228	6)·1279
			Mean ·0371	Mean ·00213

Co-efficient = 1 + (4 × ·0371) + ·00213.

```
 ·0371
     4
 ─────
 ·1484
 ·00213
 ──────
 ·15053
 1·00000
 ───────
 1·15053 = co-efficient of augmented service.
```

Mean of the girths = ...	22·78
× length on load line	105
	113·90
	22·78
	2391·9
× co-efficient augmentation ...	1·15
	119·595
	239·19
	2391·9
	2750·685 = augmented surface in square feet.

The indicated horse-power on the trial trip was 240, which, multiplied by 20,000, the co-efficient of propulsion for clean painted iron, and divided by augmented surface, the quotient will be the cube of the speed.

$$\frac{240 \times 20{,}000}{2750{\cdot}68} = 1745 = \text{cube of speed} = 12{\cdot}04$$

According to this rule, the Xantha should have attained a speed of 12·04 knots per hour on her trial: the actual speed made was 11·8 knots, but the discrepancy would have been greater at much higher or lower speeds.

To ascertain the indicated horse-power that it will take to drive a steam vessel a given speed, the calculation will be thus made. Let the given speed be 12·04 knots, the cube of which is 1745; then 1745, multiplied by the augmented surface and divided by 20,000, will in the quotient give the required power:

$$\frac{1745 \times 2750{\cdot}68}{20{,}000} = 240 \text{ I.H.P.}$$

Professor Rankine found that the computation of speed under his rule might frequently be falsified by results, mainly owing, when the speed attained fell short of that computed, to defects either in shortness or fulness of the after body, and he formulated a correction to be added to the augmented surface in the case of such defects. But the formula for he correction, as it stands, gives anomalous results.

For computing the resistance in pounds avoirdupois, Professor Rankine supplied the following rule: *Multiply the augmented surface by the square of the speed in knots, and divide the products by* 100; the quotient will be the resistance in pounds. Applying this rule to the Xantha, we have

$$\frac{2750 \times (11{\cdot}8)^2}{100} = 3822\text{lb.}$$

This rule is based on the assumption that, at speeds of 10 knots, the resistance is 1lb. per square foot of augmented surface, and that at other speeds it varies as the square of the speed.

The general conclusion with regard to this rule is that for speeds up to the wave-making period it would be safer to take the actual immersed surface. The surface per I.H.P. varies from 3 sq. ft. to 20 sq. ft. in yachts, according to the I.H.P.

In making comparisons for speed, it would always be better to first ascertain the "corresponding speeds" of the vessels concerned (see page 164). Say a yacht is to be built of similar form and of dimensions proportional to Oriental, whose speed can be fixed at 11 knots. The water-line length of Oriental is 124·3ft., and that of the assumed vessel can be put at 132ft., and her corresponding speed will be

$$\sqrt{\frac{132}{124 \cdot 3}} \times 11 = 11 \cdot 34 \text{ knots.}$$

It took 300 I.H.P. to drive the yacht 11 knots, and according to the "corresponding speed" theory it would take 366 I.H.P. to drive the larger vessel 11·34 knots, thus:

$$\left(\frac{132}{124 \cdot 3}\right)^3 \times 306 = 366 \text{ I.H.P.}$$

By the formula given on pages 293 and 297 (using Oriental's co-efficient of 180 for 11 knots) the I.H.P. for the enlarged Oriental would be 376, a sufficiently near approximation for such speeds and closeness in size. The difference in the results is consequent on the fact that in the common speed formula the fact is not provided for that for equal speeds the larger "similar ship" is experiencing a smaller ratio in the growth of resistance than the other is; and at the corresponding speed only does the ratio become equal.

If it were sought to build a yacht very much smaller than Oriental, say 80ft. long, but on her lines, the speed corresponding to Oriental's 11 knots would be

$$\frac{11}{\sqrt{\left(\frac{124 \cdot 3}{80}\right)}} = 8 \cdot 9 \text{ knots.}$$

The I.H.P. required to drive this smaller Oriental 8·9 knots would be

$$\frac{306}{\left(\frac{124 \cdot 4}{80}\right)^3} = 92 \text{ I.H.P.}$$

An example of the general correctness of this method of comparison is afforded by the Primrose (although she is not exactly similar in form to Oriental) 77ft. on the water-line, and 67 tons displacement, she having attained a speed of 8·9 knots with 93 I.H.P.

These comparisons, it should be noted, are made when the resistance is growing in a faster ratio than the cube of the speed, and if it were sought to ascertain what I.H.P. it would take to drive the 80ft. yacht to a higher speed—say 11 knots—an approximate estimate would be derived from

$$\frac{11^4 \times \text{I.H.P.}}{8{\cdot}9^4} = \text{I.H.P.} = \frac{14641 \times 93}{6274} = 220 \text{ I.H.P.}$$

This estimate is probably a little over the mark, but it serves to show the enormous expenditure of power required to overcome the resistance shortly after the wave-making period commences.

This is particularly noticeable in comparing small vessels with large, as when the former are commencing to feel sensible resistance from wave-making, the large vessels would be experiencing frictional resistance only. A comparison cau be made with the following three yachts:

	Mazeppa.	Oriental.	Primrose.
Length of water line	148ft.	124ft.	77ft.
Breadth	22ft.	20·1ft.	15ft. 5in.
Draught	10ft. 2in.	10ft. 2in.	6ft.
Displacement	316 tons.	266 tons.	67 tons.
Wetted surface	3785 sq. ft.	2755 sq. ft.	1084 sq. ft.
I.H.P. for speed of 5 knots	40	24	15
" " 6 "	55	35	24
" " 7 "	76	54	37
" " 8 "	100	82	56
" " 9 "	140	126	101
" " 10 "	196	196	152
" " 11 "	306	306	220
" " 11·5 "	375	375	—
I.H.P. per 100ft. wet surface at 5 "	1·05	0·90	1·40
" " 6 "	1·40	1·20	2·20
" " 8 "	2·60	3·00	5·60
" " 10 "	5·10	7·13	14·00

No doubt at six knots Primrose would already be feeling some slight resistance from wave-making, and at the lower speed her frictional resistance would be somewhat in excess of that of the other two from causes explained on page 119.

At six knots the Oriental and Mazeppa have about equal I.H.P. per 100ft. of surface, but after passing that speed it will be seen that Oriental required a much greater allowance; in short, with 27 per cent. less area of immersed surface she took exactly the same horse power as Mazeppa at 10, 11, and 11·5 knots. This will show the unreliability of the $\frac{\text{I.H.P.}}{\text{Wet surface}}$ test of efficiency, except for speeds when there is absolutely no wave making.

Most of the speed formulæ which have been dealt with are open to the objection that they assume in one common term that the efficiency of

the ship, the engine, and the propeller have a constant value, and that the resistance must be as the square of the speed and the I.H.P. as the cube. It has already been pointed out that the growth of the resistance does not vary uniformly as the square of the speed, and it remains to be shown that the speed formulæ may make it appear to grow in a much larger ratio than it really does at high speeds, owing to the varying efficiency of the engine and of the propeller in relation to its frictional resistance and slip.

As already said, it has been assumed that the I.H.P. is a force of uniform effective value: that is to say, that the power utilised in overcoming a vessel's resistance always bears the same relation to the work done in the cylinders, no matter what the description of engine. This, however, is by no means the true state of the case, and the useful work done in overcoming a vessel's resistance varies very considerably in relation to the total work done by the machinery. The late Mr. Froude estimated that not more than from 38 to 40 per cent. of the indicated horse power was employed in propelling a vessel.*

This conclusion, however, was arrived at some time ago from investigations carried on with very large ships and engines, and there is no doubt that, so far as the most modern yachts are concerned, a very much larger percentage of the I.H.P. is utilised, reaching even higher than 60 per cent. of the total in well designed and engined steam yachts.

Mr. Isherwood, of the United States Navy, made some experimental trials in this respect with the yacht Lookout, of about 100 tons O.M., and the results were published in the Journal of the Franklin Institute in 1881. The experiments were mainly made to ascertain the effect of different propellers on the efficiency of the engines, and if the method of analysing the results was not quite satisfactory, they are to a large extent instructive.

The dimensions, &c., of the Lookout were as follows:

HULL.

Length on water line to aft side sternpost	96ft.
Breadth on water line	13ft. 6in.
Extreme breadth	16ft.
Extreme draught of water aft	4ft. 7in.
Area of mid-section	25·28 sq. ft.
Displacement	42·87 tons.

ENGINE.

Diameter of the small cylinder	12in.
Diameter of the large cylinder	20in.
Stroke	16in.

* *Vide* Transactions of the Institution of Naval Architects, 1876.

BOILER.

Diameter	7ft.
Length	8ft.
Diameter of the furnaces (two)	2ft. 4in.
Length of the grates	5ft.
Area of grate surface	23½ sq. ft.
Heating surface	519 sq. ft.
Cross area of all the tubes	3·14 sq. ft.
State room	94·54 cu. ft.

SCREW.

Designation of the screws.	Diameter in feet.	Pitch in feet.	No. of blades.	Greatest length of screw in feet in direction of axis.	Area of blade surface in square feet.
A	4·6458	8·400	4	0·9167	7·9447
B	5·0000	7·500	4	1·1875	13·5266
D	5·0000	8·625	4	0·4461	5·3470
E	5·0000	9·000	4	0.8125	9·3407

RESULTS OF EXPERIMENTS.

		Screw.	A	B	D	E*
		Speed of the vessel per hour, in knots	9·5341	10·0974	10·4190	10·1929
		Slip of the screw, in per cent. of its speed	24·4072	13·6772	20·1542	17·9739
		Thrust of the screw, in pounds	1650·9925	1851·0375	1970·5470	1909·3913
		Revolutions of the screw per minute	152·3085	158·1988	153·4601	140·0500
		Indicated pressure on the piston, in pounds per square inch	21·9103	23·0038	25·8904	26·8090
Distribution of the indicated horse power.	Absolute.	Indicated horse power	84·4804	92·7090	100·5812	95·0489
		Horse power expended in working the engine, *per se*	10·4737	10·9475	10·5529	9·6307
		Net horse power applied to the crankpin	74·0067	81·7615	90·0283	85·4182
		Horse power absorbed by the friction of the load	5·5505	6·1321	6·7521	6·4064
		Horse power expended in overcoming the resistance of the water to the surface of the screw blades	4·4483	9·0767	4·2398	6·0816
		Horse power expended in the slip of the screw	15·6225	9·1025	15·9291	13·1084
		Horse power expended in the propulsion of the vessel	48·3854	57·4502	63·1073	59·8218
	Proportional.	Per cent. of the net horse power applied to the crankpin, absorbed by the friction of the load	7·5000	7·5000	7·5000	7·5000
		Per cent. of the net horse power applied to the crankpin, expended in overcoming the resistance of the water to the surface of the screw blades.	6·0107	11·1014	4·7094	7·1198
		Per cent. of the net horse power applied to the crankpin, expended in the slip of the screw	21·1096	11·1330	17·6934	15·3462
		Per cent. of the net horse power applied to the crankpin, expended in the propulsion of the vessel	65·3797	70·2656	70·0972	70·0340

* It should be noted that these experiments were made with an uncoppered hull; the hull was subsequently coppered, and an experiment with the E propeller showed some advantage in point of resistance, but the limited extent of the gain shows that the uncoppered hull must have had a very good surface.

DISTRIBUTION OF THE POWER WITH THE HULL COPPERED.

	Horse power.		Per cent. of net horse power.
Indicated power developed by the engine	79·5393		
Power expended in working the engine, *per se*	9·2265		
Net power applied to the crankpin	70·3128	or	100·0000
Power absorbed by the friction of the load	5·2734	„	7·5000
Power expended in overcoming the resistance of the water to the surface of the screw blades	5·3474	„	7·6051
Power expended in the slip of the screw	9 5507	„	13·5832
Power expended in the propulsion of the vessel	50·1413	„	71·3117
Totals	70·3128	or	100·0000

From the foregoing it will be seen that Mr. Isherwood estimates 70 per cent. of the work done by the engines was utilised in propelling the vessel, and although his distribution of the power may be somewhat too favourable for the engine, it is not far out. The different screws do not appear to have much affected the efficiency of the engines, the A screw showing the greatest variation.

The late Mr. Froude devised a method of separating the engine power taken up by the propeller, from the gross I.H.P., and termed the result "Indicated Thrust." His rule for estimating this thrust is the mean piston pressure multiplied by the piston travel per revolution, and divided by the pitch of the propeller. "This is the thrust which the propeller would be exerting if the force of the steam were employed wholly in creating thrust instead of partly overcoming friction, driving the air pump, and overcoming other collateral resistances."

As the mean piston pressure is not always obtainable, the rule was converted to

$$\text{Indicated thrust} = \frac{33{,}000 \times \text{I.H.P.}}{\text{Pitch} \times \text{Revolutions.}}$$

This indicated thrust is taken as equal to the effective horse power (E.H.P.) absorbed by the resistance of the vessel; and of this E.H.P. it was estimated by Mr. Froude that 0·4 of it went to overcome the augmented resistance due to the action of the propeller in diminishing pressure at the stern (see pp. 129 and 291); and 0·1 of the E.H.P. to overcome the water friction on the screw blades. 0·4 is probably much too high a percentage of the E.H.P. to set down for augmented resistance for most yachts, and probably 0·2 would be a nearer approximation of the truth.

It is quite plain from a study of the table on page 304 that there must be a point low down in the I.H.P. curve, as shown in Fig. 138, where the friction of the engine due to the working load would prevent any real thrust being exerted. In constructing thrust curves similarly to the I.H.P. curves this became apparent; and the results of some trials made

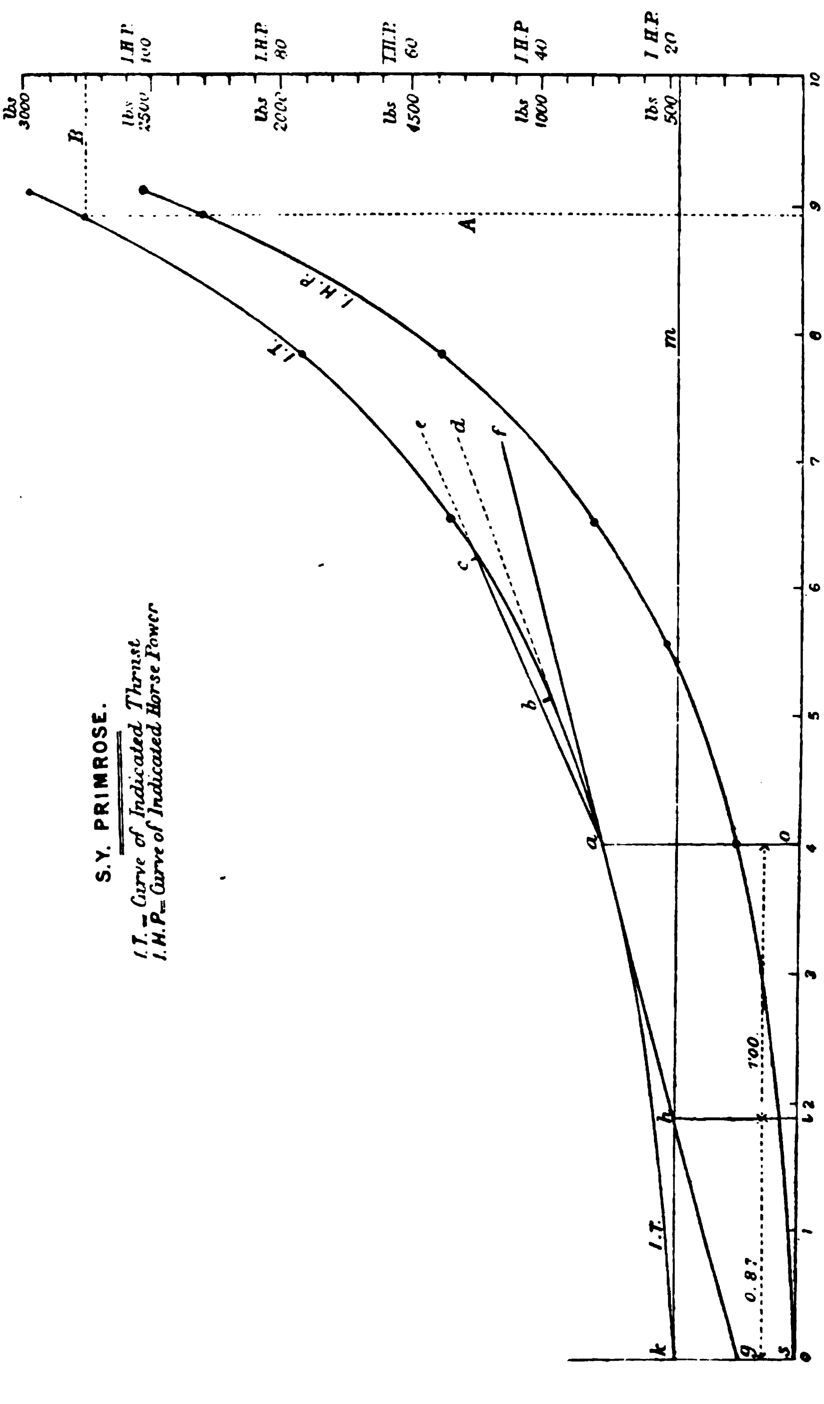

S.Y. PRIMROSE.
I.T. = Curve of Indicated Thrust
I.H.P. = Curve of Indicated Horse Power
I.T.
I.H.P.
A
B
m

FIG. 188.

by Messrs. Denny, of Dumbarton, led Mr. Froude to devise the following method for graphically illustrating the point in the curve where there can be no propulsive thrust. In Fig. 138 the curve I T represents the curve of indicated thrust for the yacht Primrose, calculated for different speed by the above formula. The "spots" in the curve read at right angles (see A) to the base line show the speed, and at right angles (see B) to the side perpendicular they show the indicated thrust in pounds. The curve I T, it will be seen, is calculated down to four knots, and the line *k h m* represents the constant frictional load of the engines themselves. This line of constant friction is thus found. From the point *a* cut off a parabolic segment of convenient length as to *c*; bisect *a c* (*i.e.*, divide equally) at *b*; through *a b* draw *a d*, then draw *a f*, making the angle *a d f* equal to angle *a d e*. *a f* is the tangent of the curve at the point *a*. Produce *a f* to *g*, and draw *a o* at right angles to the base line; divide *o s* in the proportion 1 : ·87 as at *i*; draw *i h*. Produce *k m* through *h* and parallel to the base line as the line of marking the part of the I.H.P. to cut off for constant friction. The line *h i* indicates the point in the curve when there could be no thrust or speed. See also diagram for Cressida, page 299.

The I.H.P. curve is put in merely to compare its character with the I T curve; and it will be noted that its power scale is different from the I T scale.

The I T curve of Primrose shows a rapid gain in thrust as the speed advances from five to nine knots; but the I.H.P. curve is growing still faster, and shows that there must be some loss due to increased propeller slip and decreased engine efficiency as the number of the revolutions increase. Still, the curves are close in character, and compare favourably with the thrust and I.H.P. curves for larger yachts and engines. It will be noted that the I.H.P. and I T curves of the steam yacht Cressida, shown on page 299, are very similar in character to the much smaller vessel Primrose, and this similarity will generally be found to exist.

Mr. Froude resolved "indicated thrust" into its several elements as follows: "1. Useful thrust equal to the ship's true resistance. 2. The augment of the resistance due to the action of the propeller on the closing in water around the stern. 3. Friction of the water on the blades of the propeller. 4. Friction of the engine parts. 5. Increased friction of the engine parts due to its load. 6. Air and feed pump work."

"2, 3, and 4 are nearly proportional to 1 (the useful thrust); 6 to the square of the revolutions; 5 constant at all speeds."

For further comparison the following tables concerning some well-known steam yachts can be referred to.

Speed in knots.	Speed of screw in knots.	Revolutions.	Slip per cent.	I.H.P.	Indicated thrust in pounds.	Power of speed to which I.H.P. is proportional.	Power of speed to which indicated thrust is proportional.
PRIMROSE.							
5·5	6·35	103	13·3	19·4	1002	——	——
7·8	9·25	150	15·7	53·3	1890	Cube	Square
8·9	11·16	181	20·2	93·0	2735	4th power	2·8 power.
ORIENTAL.							
7	7·93	80·5	12	53	2172	——	——
8·3	9·57	97	13	95	3232	2·5 power	Square
11·2	14·40	146	22·2	330	7459	4th power	2·8 power
MAZEPPA.							
6	7·10	60	15·5	55	2563	——	——
8	9·47	80	15·5	100	3437	Square	As simple speed
10	11·84	100	15·5	200	5500	Cube	Square
11·5	14·20	120	19·0	375	8250	4·4 power	2·9 power
MALIKAH.							
7·4	7·69	60	3·9	66	2792	——	——
9·5	10·20	80	7·0	163	5174	Cube	Square
10·2	11·50	90	11·3	204	5980	Cube	Square
11·2	12·80	100	12·5	280	7107	Cube	Square
12·1	13·50	106	11·0	361	8646	4th power	Cube
12·4	14.70	114·5	15·6	441	9790	5th power	4th power

	PRIMROSE.	ORIENTAL.	MAZEPPA	MALIKAH.
Length on water line	77ft.	124·4ft.	148ft.	144ft.
Beam	15·5ft.	20·1ft.	22ft.	21·5ft.
Draught of water	6ft.	10·2ft.	10·2ft.	10·3ft.
Displacement	67 tons.	266 tons.	316 tons.	328 tons.
Area of immersed surface	1084 sq. ft.	2755 sq. ft.	3785 sq. ft.	—
Diameter of propeller	4·6ft.	7ft.	—	8·75ft.
Pitch of propeller	6·25ft.	10ft.	12ft.	13ft.
Surface of all blades	6·25 sq. ft.	13·5 sq. ft.	—	23·2 sq. ft.
Number of blades	4	—	—	4

The value of measured mile trials is often questioned, but there can be no doubt that without such trials the information available for experimental purposes would not be sufficiently definite or exact. In the Royal Navy it is the practice to make three runs with the tide and three against and take the mean of the six runs as the true speed or the "mean of means." Thus, to get sufficient speeds to make a progressive trial, a great many runs have to be made, and the time occupied will be considerable.

However, providing a time be chosen at about slack water, better results can be obtained by a pair of runs than from three pairs in tidal streams of unequal velocity, as will appear from the results of the trial trip of the steam yacht Celia, given below. But even if the current were of uniform velocity, it might still slightly affect the I.H.P. and the speed performance, as it is an observed fact in the numberless measured

mile trials of the ships of the Royal Navy, that when running with the stream, a sensibly greater number of revolutions are made for the same boiler pressure than when running against stream. This can only be accounted for by the fact that a tidal stream always has a gradient in the direction of its course, although it may only be $\frac{1}{20000}$, and this would tend to give the vessel, by her own weight, motion through the water, and thus help the screw, small though it would be; and the growth in the resistance would prevent any speed of the vessel due to the gradient going on increasing indefinitely.

Measured Mile Trials of the Steam Yacht Celia.

Runs.	Knots.	1st means.	2nd means.	3rd means.	4th means.	Mean of means.
1.	9·19	9·74	9.69	9·68	9·64	9·63 true speed.
2.	10.28	9·64	9·67	9·61	9·62	
3.	9·01	9·69	9·56	9·60		
4.	10·36	9·42	9·63			
5.	8·48	9·83				
6.	11·19					
	6)58·51		4)38·55			
	9·75 ordinary mean speed.		9·64 ordinary mean of second means.			

It will be seen that the 5th and 6th runs were made as the current began to grow in strength, and in reality the first four runs would give the truest result, as follows :

Runs.				
1.	9·19	9·74	9·69	9·68 true mean speed.
2.	10·28	9·64	9·67	
3.	9·01	9·69		
4.	10·36			
	4)38·84			
	9·71 ordinary mean speed.			

In ordinary practice, if the trial is made in slack water, the mean of two runs would be taken as sufficiently accurate.

Prior to going on the mile, the fires should be got into good condition, so as to obviate the necessity of stoking, and a steady head of steam should be maintained all through. Also the revolutions should be counted over four or more minutes, according to the duration of the run and the mean taken for calculating the I.H.P. A common practice is to go on the mile when the stokers have got a good head of steam; and then to admit full steam into the cylinders, diagrams being meanwhile taken, and the revolutions counted. If the boiler is not capable of continuously making the steam which the cylinders will take, the pressure drops, there is then, of course, a smaller I.H.P., and the last part of the mile is done at a much slower rate than the first portion. This may not matter, so far as the I.H.P. is concerned, and the makers of the engines may consequently be satisfied; but not so the builder or designer of the vessel, as the result will apparently indicate that the vessel is a bad one to drive.

If the contract with the builders of the engine has been for so much I.H.P., the result might also be influenced by the smallness of pitch and surface of the propeller, as high piston speed means high I.H.P., as calculated from the diagrams and number of revolutions, but it may also mean poor efficiency so far as the engines are concerned. In such a case, the inefficiency of the engine and propeller would be discovered by a curve of indicated thrust. The I.H.P. would be found to increase to a very high power of the speed, whilst the thrust increased at a low power; or, say the I.H.P. increased as the fourth power of the speed, the thrust might only increase as the square (see page 307, &c.). Some loss of thrust will, however, always appear at the higher speeds, as the slip increases with the increase in the revolutions.

If the engineers design and build the vessel as well as the engines, a certain speed might be contracted for; but it is essential that proper precautions should be taken to insure that the whole of the machinery is suitable for sustaining the given speed for any length of time, and economically, and not by means of a small engine with a large boiler and grate area, perhaps farther assisted by the steam blast.

If the contract be for so many revolutions of a propeller of a certain diameter, pitch, blade, area, &c., the trial should be of three or four hours duration; and it would be also necessary, as in the other case, to stipulate for a boiler and machinery likely to do the work efficiently and economically.

The term "auxiliary steam yacht" seems almost a misnomer at the present time, as the steam power put in is usually capable of driving the yacht at a speed at least four-fifths of that of a yacht of similar size which is provided with "full steam power." In the case of full rigged yachts which can steam from nine to eleven knots an hour, it may at times be a question whether the use of steam will increase the speed; or, if the yacht be already under steam, if setting her sail will increase her speed, and to what extent. It would be impossible to make a precise computation of this, as the propulsive power of sail is not amenable to calculation, but the following table has been compiled from observed results as an approximation:

Ratio of speed under sail alone to speed under steam alone.	Then the probable speed under sail and steam together will bear the following ratio to speed under steam.
0·5	1·04
0·7	1·10
1·0	1·25
1·3	1·47
1·5	1·64
1·7	1·81
2·0	2·08

According to the foregoing, if a yacht can steam at the rate of, say, nine miles an hour, and if there is a beam wind which will enable her to

sail at that rate without steam, then her speed would be increased one-fourth by the use of steam and sail together, or $9 \times 1{\cdot}25 = 11{\cdot}25$ knots. The increase of speed would, of course, depend upon the growth of the resistance of the yacht at the assumed speed of nine knots.

Or, suppose the speed under sail to be ten knots, and the speed under steam to be five knots, then the speed under sail and steam combined will be $5 \times 2{\cdot}08 = 10{\cdot}40$ knots. This shows that the application of what originally was understood as auxiliary power was of very little use in increasing the speed, except in very light winds, when the speed under sail would be low, say five knots, then the gain would be one-fourth, or $5 \times 1{\cdot}25 = 6{\cdot}25$ knots.

When steam yachts are designed for sailing without the aid of their steam power, it is an advantage to have them fitted with "feathering screws"—that is, screws whose blades can be turned in the boss, so that they come edgeways in the line of motion. Various methods for feathering have been patented, but the one in most general use is that known as Bevis's.

If the propeller does not feather, it should be made to lift or disconnect, so as to revolve freely. A plan for lifting the screw was largely adopted in the early days of screw propulsion, but it is seldom resorted to now. But, even with the propeller disconnected and rotating freely, it adds to the resistance very considerably, estimated by Mr. Froude at 0·1 of the ship's total resistance at ten knots in the case of H.M.S. Greyhound;* and this is borne out by the experiments made by Mr. Isherwood, described on page 306. If the blades are simply feathered they will not offer so much resistance, as the friction from them will not be so great as when rotating; and, on the whole, this is the most satisfactory arrangement at present in use to get over the difficulty of the propeller when under sail alone. If the propeller has only two blades it should be kept parallel to the sternpost, not across it; and if there are four blades, two of them should be so arranged. The engineer must, of course, know the position of the blades in relation to the shaft crank.

In designing a steam yacht the naval architect must exercise great care in providing sufficient displacement for the coal it is considered desirable to carry, and also, in the case of an auxiliary steam yacht, of the ballast. As a rule the latter have more beam than the ordinary steam yacht, and greater under water depth, and they are consequently steadier in a sea, providing they are not over sparred. At low speeds they are as economical to drive, and their chief advantage is for ocean voyages, and they have achieved some very remarkable runs.

* *Vide* "Transactions of Institution of Naval Architects," 1876.

The Lancashire Witch, entirely under canvas, did the distance (4458 miles) from the Falkland Islands to Natal in twenty-three days, the greatest run being 259 miles; and she did the distance from Yokohama (4400) also in twenty-three days, her greatest run being again 295 miles in twenty-four hours; and she did the whole distance from Tahiti to Liverpool (11,030 miles) under canvas, in seventy-nine days, having covered, owing to head winds, 12,230 miles. The Sunbeam, in her voyage round the world, covered 35,450 miles, and out of this distance 20,400 were done under canvas, without steam; she also, like Lancashire Witch, did many good runs under canvas, having on one occasion logged 299 miles. The Sunbeam's best day's steaming was 230 miles, and Lancashire Witch's 216.

The advantage of an auxiliary may be therefore very considerable in extended voyages where the trade winds can be made use of, and even in head winds if she be a fairly weatherly vessel.

In the case of either an auxiliary or a full steam yacht, it is of importance that the coal should be carried amidships; and often to effect this a large athwart-ship bunker is provided amidship in order that the coal may be used from about the centre of buoyancy. Occasionally bunkers are provided under the cabin floor to help trim the vessel, and this is often necessary when the engines are placed abaft the centre of buoyancy.

In apportioning power to an auxiliary steam yacht, it is important that the coal consumption should be considered, as putting in enough power to obtain another knot an hour speed may entail a serious demand on the internal space of the yacht and an addition to the daily consumption, which would necessitate either an increase in the size of the bunkers if long voyages had to be made, or frequent coaling for coast work if the bunkers were of small capacity. An auxiliary steam yacht of 350 tons would be supplied with machinery capable of developing 300 I.H.P., and that power would drive her at the rate of about 10 knots an hour. With ordinary two-cylinder engines and a working boiler pressure of 100lb., the coal consumption would be about 1·8lb. per I.H.P. per hour. The bunker capacity would be about 80 tons, and if the yacht were driven at 10 knots the coal would last fourteen days, and the yacht would cover 3360 miles in that time.

It would, however, be more likely that the yacht would not be driven faster than 9 knots, which would require about 218 I.H.P. In this case the coal would last nineteen days and enable the yacht to cover 4100 miles. In this case the cost per mile in coal would be 45lb.; but if the speed had been 10 knots 53lb. of coal would have been used for each mile travelled. At 8 knots the yacht would steam 4950 miles in 25 days 15 hours, at a cost of 36lb. of coal per mile steamed. These figures have been proved in actual practice.

The saving in coal by using steam of 160lb. pressure as against steam of 100lb. is from 15 to 20 per cent., or, say, about one sixth. The steam in this case would be in three cylinders, according to what is popularly called the triple expansion system. If the auxiliary yacht were fitted with such engines of 300 I.H.P., and driven at 10 knots, her 80 tons of coal would last 16 days 8 hours, and she would cover 3920 miles in that time. If driven at nine knots the coal would last 22 days 4 hours, and the yacht would cover 4800 miles in that time. The cost per mile in coal at 9 knots would be 37·3lb., and at 10 knots, 45lb. per mile, at 8 knots an hour the yacht could steam 5775 miles on a consumption of coal of 30lb. per mile covered, the time occupied in traversing the distance being thirty days. The triple expansion engine therefore shows to great advantage for ocean steaming, and the actual saving in coal bill expenses might be considerable in out of the way places where coal is 2*l.* or 3*l.* per ton.

In a voyage made by the auxiliary steam yacht Marchesa to South America and the West Indies in 1878, careful coal consumption tests were made. The working pressure was 70lb., and the greatest I.H.P. 230, which gave about 9·3 knots an hour. The bunker capacity was 80 tons, and the results of the observations made were as follows:

Coal burnt per 24 hours.	Speed per hour.	Coal would last.	Total miles steamed.	Coal per mile.
3 tons.	8 knots.	26 days 14 hours.	5104	35lb.
4 tons.	8¾ knots.	20 days.	4200	43lb.
5 tons.	9¼ knots.	16 days.	3552	51lb.

Thus, by burning 3 tons of coal per day instead of 5 tons, the Marchesa could steam 1552 miles farther, and would only be ten and a half days longer over the greater distance. It will thus be seen that great knowledge and judgment are necessary in designing and providing an auxiliary yacht with steam power, and in using it. But it is no less important in a full power steamer, as, though the alteration of trim due to coal consumption may not be such a serious matter for her; yet it may happen to be of the greatest importance that the economical speed should be known, if the yacht is short of coal and far from a coaling station. It may be thought that the proper course would be to drive the yacht as fast as possible, and reach the coaling place in the shortest time; but this may only result in the yacht being unable to reach her port at all.

A modern fast steam yacht of about 500 tons and 600 I.H.P. would have an average speed of about 13 knots, and, say, a bunker capacity of

100 tons; and, with the daily consumption set forth in the tables, she would steam at the following rate and distances:

Coal burnt per 24 hours.	Speed per hour.	Coal would last.	Total miles steamed.	Coal per mile.
3 tons.	8 knots.	33 days 6 hours.	6394	35lb.
4½ tons.	10 knots.	22 days 2 hours.	5328	42lb.
10 tons.	13 knots.	10 days 2 hours.	3235	70lb.

This simply means that such (leaving stress of weather out of the question) could, at 8 knots an hour, steam to New York and back on 100 tons of coal; but if the attempt were made to reach there in about ten days, her coal would only just last the outward voyage.

The Wanderer steam yacht, 700 tons (now re-named Vagus, see the table on page 319), in her voyage round the world, burnt 66lb. of coal per mile steamed. The Maid of Honour (see page 318), at an average speed of 10·8 knots an hour, burnt 32lb. of coal per knot, steamed to and from the North Cape.

A fast steam yacht of about 100 tons, such as Linotte (particulars of which will be found on pages 290 and 318), consumes about 30lb. of coal per mile travelled at the rate of 12 knots an hour; 25lb. per mile at 11 knots, and 22lb. per mile at 10 knots, and 19lb. per mile at 9 knots.

The first cost of a 350-ton auxiliary steam yacht would exceed that of the cost of a full steam yacht, as, although the engines might not cost quite so much, the heavy spars, rigging, and sails would cost probably ten times as much as those in the steamer.

The cost of working a 350-ton auxiliary steam yacht, so far as engine-room wages go, would be about the same as the full steam yacht; but there would be six additional seamen to maintain. There would be also wear and tear of sails; but, as the engine and boiler would probably be used less in proportion, a fair set-off would be arrived at. However, for home cruising, or even for cruising to the Mediterranean or Baltic, the full power steam yacht will prove the more economical and satisfactory.

Great care should be always taken in selecting an engineer; he should not only know how to drive an engine, but be capable of readjusting any of its parts and effecting slight repairs. The second engineer should be what is known as a "donkey man."

A "certificated second engineer" is frequently engaged to take charge of a yacht's engines, and, if a steady, sober man can be found, a "certificated second engineer" will be a most valuable one to employ. These second engineers have often been "donkey men," that is, men who, having acquired a fair education, have had experience in the shop, and have

TABLE OF ELEMENTS OF STEAM YACHTS.

Lloyd's yacht tons.	Maid of Honour. 182	Fauvette. 420	Samara. 108	Paulina. 317	Alabama. 76	Vista. 95	Sea Horse. 106	Queen Mab. 114	Speedy. 140	Monsoon. 232	Lantana. 225
Length on L. W. L. to outer sternpost	117ft.	160ft.	95ft.	144·5ft.	88ft. 4in.	92ft.	108ft.	100ft. 3in.	108ft.	129ft.	124ft.
Breadth extreme	17ft. 9in.	22ft. 8in.	16ft.	21·2ft.	14ft.	14ft. 6in.	14ft. 7in.	13ft.	16ft. 2in.	19ft. 1in.	19ft. 1in.
Draught of water aft	11ft. 1in.	12ft. 11in.	6ft. 10in.	10ft. 7in.	4ft. 4in.	6ft. 2in.	6ft. 6in.	8ft. 9in.	9ft. 6in.	10ft. 4in.	10ft. 4in.
Area of midship section...	109 sq. ft.	181 sq. ft.	94 sq. ft.	142 sq. ft.	—	58 sq. ft.	62·5 sq. ft.	—	96·2 sq. ft.	—	—
Displacement on trial......	298 tons.	491 tons.	149 tons.	343 tons.	54 tons.	101 tons.	112 tons.	132 tons.	178 tons.	280 tons.	261 tons.
Area of wet surface	2710 sq. ft.	4590 sq. ft.	—	361 sq. ft.	—	—	—	—	2290 sq. ft.	—	—
Ballast..	33 tons.	45 tons.	7 tons.	—	—	—	13 tons.	15 tons.	15 tons.	30 tons	36 tons.
Reg. tons gross and net	137 & 94	330 & 126	—	264 & 123	76 & 53	72 & 49	86 & 58	87 & 59	113 & 77	171 & 116	164 & 112
Fore-and-aft engine and boiler space	—	—	—	—	—	21ft.	25ft.	—	—	—	—
Diameter of cylinders......	15½in. & 30in.	18in., 30in., 48in.	10, 16, 25in.	20in. & 41in,	7, 10½, 16	12in. & 25in.	9½, 15, 25	10½in. & 20in.	12, 19, 30	15½in. & 30in.	15½in. & 30in.
Stroke of pistons	23in.	32in.	18in.	24in	12in.	18in.	16in.	18in.	21in.	23in.	23in.
Revolutions	114	101	148	105	240	154	197	123	138	110	110
I.H.P. on trial.................	294	1150	255	612	130	191	270	118	395†	266	270
Vacuum	—	—	—	—	24in.	27in.	26in.	27in.	—	25in.	—
Diameter of boiler	9ft. 10in.	11ft. 10in.*	8ft. 9in.	12ft.	—	8ft.	8ft. 9in.	8ft. 6in.	10ft. 6in.	10ft.	9ft. 10in.
Length of boiler	8ft. 9in.	9ft. 4in.*	9ft.	10ft. 3in.	—	7ft. 10in.	9ft.	8ft.	9ft.	9ft.	8ft. 9in.
Heating surface..............	818 sq. ft.	2300 sq. ft.	709 sq. ft.	1362 sq. ft.	—	555 sq. ft.	605 sq. ft.	—	1003 sq. ft.	856 sq. ft.	818 sq. ft.
Grate surface.................	36 sq. ft.	84 sq. ft.	32 sq. ft.	54 sq. ft.	—	25 sq. ft.	22 sq. ft.	—	36 sq. ft.	38 sq. ft.	36 sq. ft.
Working pressure	100lb.	160lb.	160lb.	110lb.	165lb.	—	160lb.	90lb.	180lb.	100lb.	100lb.
Total weight of machinery with steam up	43 tons.	167 tons.	43 tons.	—	—	—	—	—	59 tons.	—	43 tons.
Diameter of propeller......	8ft.	10ft. 9½in.	6ft.	8ft. 9in.	—	5ft. 6in.	5ft. 9in.	6ft. 3in.	7ft. 9in.	8ft.	8ft.
Pitch.............................	12ft. 6in.	17ft. 8in.	10ft.	14ft.	5ft. 10in.	9ft.	7ft. 6in.	9ft.	10ft. 10in.	12ft. 6in.	12ft. 6in.
Surface of all the blades...	17 sq. ft.	45 sq. ft	12 sq. ft.	24 sq. ft.	—	11 sq. ft.	9·5 sq. ft.	12 sq. ft.	16 sq. ft.	18 sq. ft.	18 sq. ft.
Number of blades	4	4	4	4	—	4	4	4	4	4	4
Speed on trial.................	12·21 knots.	15·6 knots.	11·1 knots.	12·7 knots.	11 knots.	11·2 knots.	12·3 knots.	9·5 knots.	13·1 knots.	10·1 knots.	11 knots.

* Two boilers and forced draught. † With forced draught; 280 natural draught.

TABLE OF ELEMENTS OF STEAM YACHTS.

Lloyd's yacht tons.	Imogen. 437	Fire Fly. 402	Mazeppa. 370	Malikah. 328	Adriana. 322	Blonnag-na-mara. 310	Mera. 293	Queen Marfisa. 160	Erne. 158	Linotte. 90	Tern. 64
Length on L. W L. to outer sternpost....................	155ft.	170ft.	148ft.	144ft.	148ft. 6in.	144ft. 9in.	136ft.	112ft.	113ft.	88ft. 3in.	68ft.
Breadth extreme	24ft.	22ft.	22ft.	21ft. 6in.	21ft.	21ft.	21ft.	17ft.	17ft.	14ft. 6in.	14ft. 6in.
Draught of water aft	11ft. 5in.	10ft. 6in.	10ft. 2in.	10ft. 3in.	11ft. 1in.	12ft. 5in.	10ft. 4in.	10ft.	10ft. 3in.	6ft. 4in.	10ft.
Area of midship section...	—	—	119 sq. ft.	134 sq. ft.	129 sq. ft.	129 sq. ft.	118 sq. ft.	101 sq. ft.	95 sq. ft.	54 sq. ft.	71 sq. ft.
Displacement on trial......	447 tons.	301 tons.	316 tons.	328 tons.	335 tons.	343 tons.	280 tons.	185 tons.	188 tons.	84 tons.	85 tons.
Area of wet surface.........	—	—	3785 sq. ft.	—	—	3713 sq. ft.	3309 sq. ft.	2480 sq. ft.	2330 sq. ft.	1394 sq. ft.	—
Ballast...........................	—	—	—	—	—	—	29 tons.	25 tons.	—	6 tons.	23 tons.
Reg. tons gross and net...	331 & 123	247 & 118	219 & 126	220 & 110	228 & 98	241 & 69	211 & 132	127 & 87	122 & 83	—	47 & 28
Fore-and-aft engine and boiler space.................	—	49ft.	—	—	—	42ft.	29ft. 6in.	24ft. 6in.	26ft.	20ft. 4in.	14ft.
Diameter of cylinders......	17, 25, 34, 50	15, 30, 42	20in. & 38in.	15, 24, 39	11, 18½, 30	7, 16, 22, 34	18 & 35	16in. & 32in.	16in. & 30in.	12in. & 24in.	5, 8, & 13
Stroke of pistons............	33in.	24in.	24in.	24in.	24in.	24in.	24in.	18in.	18in.	18in.	10in.
Revolutions	96	128	120	115	98	113	105	110	125	165	234
I.H.P. on trial	1070	635	375	441	280	528	370	246	262	187	51
Vacuum	—	—	—	27½in.	—	—	25½in.	27¾in.	—	—	28in.
Diameter of boiler	14ft. 6in.	11ft.	12ft.	12ft.	8ft.	11ft. 6in.	10ft 8in.	9ft. 6in.	8ft. 8in,	8ft.	5ft.
Length of boiler	9ft. 6in.	15ft.	9ft. 6in.	9ft. 6in.	7ft. 10in	9ft. 10in.	8ft. 8in.	8ft. 9in,	9ft. 10in.	7ft. 10in.	5ft. 6in.
Heating surface	1800 sq. ft.	—	1057 sq. ft.	1250 sq. ft.	629 sq. ft.	—	975 sq. ft.	730 sq. ft.	804 sq. ft.	676 sq. ft.	200 sq. ft.
Grate surface..................	63 sq. ft.	68 sq. ft.	40 sq. ft.	50 sq. ft.	25 sq. ft.	40 sq. ft.	38 sq. ft.	30 sq. ft.	33 sq. ft.	25 sq. ft	75 sq. ft.
Working pressure	160lb.	150lb,	75lb.	150lb.	160lb.	130lb.	90lb.	100lb.	80lb.	100lb.	140lb.
Total weight of machinery with steam up	—	104 tons.	—	70 tons.	—	—	52 tons.	46 tons.	—	28 tons.	11 tons.
Diameter of propeller......	10ft. 2in.	9ft. 2in.	8ft. 6in.	8ft. 9in.	8ft. 9in.	10ft.	8ft. 4in.	7ft. 10in.	7ft. 6in.	5ft. 4in.	4ft. 3in.
Pitch.............................	16ft.	11ft. 10in.	12ft.	13ft.	13ft.	11ft.	13ft. 6in.	11ft. 6in.	10ft. 6in.	9ft. 6in.	4ft.
Surface of all the blades...	36·8 sq. ft.	23·6 sq. ft.	—	23·2 sq. ft.	21·7 sq. ft.	25 sq. ft.	24 sq. ft.	17 sq. ft.	—	11 sq. ft.	5 sq. ft.
Number of blades	4	4	—	4	3	4	4	4	—	4	2
Speed on trial..................	14·1 knots.	14·1 knots.	11·5 knots.	12·4 knots.	11·1 knots.	12	11·6 knots.	11·14 knits.	11·1 knots.	12·05 knots.	7·7 knots.
Co efficient $D^{\frac{2}{3}}$	153	190	187	204	235	160	182	183	171	173	174

Lloyd's yacht tons	Vagus (Wanderer). 708	Chazalle. 545	Capercailzie. 519	Gitana. 471	Jacamar. 451	Soprano (Marchesa). 377	Giralda. 301	Walrus (Queen of Palmyra). 271	Aries. 268	Sölyst (Xanthe). 256	Lady Cassandra (Oriental). 232	Sunbeam. 532	Primrose. 81
Length on L.W.L. to outer sternpost	183ft	150ft. 6in.	183ft. 10in.	153ft.	160ft. 3in.	134ft.	140ft. 9in.	128ft.	138ft. 5in.	105ft.	124ft. 4in.	154ft. 9in.	77ft.
Breadth extreme.........	29ft. 3in.	28ft. 6in.	24ft.	26ft. 1in.	24ft.	25ft.	21ft. 2in.	21ft.	20ft. 1in.	19ft.	20ft. 1¼in.	27ft. 6in.	15ft. 5in.
Draught of water aft...	15ft.	14ft. 4in.	12ft. 2in	13ft. 10in.	11ft. 6in.	13ft. 4in.	9ft. 9in.	9ft. 6in.	10ft. 2in.	9ft.	10ft. 2in.	13ft. 4in.	6ft.
Mean draught of water	—	12·3ft.	10ft. 2in.	—	10ft. 2in.	12·5ft.	9·1ft.	8·2ft.	8ft. 11in.	7·7ft.	8ft. 8in.	12ft. 7in.	5ft. 3in.
Area of L.W. plane ..	—	2908 sq. ft.	3144 sq. ft.	2808 sq. ft.	2655 sq. ft.	2360 sq. ft.	2080 sq. ft.	1780 sq. ft.	2089 sq. ft.	1520 sq. ft.	1726 sq. ft.	2917 sq. ft.	777 sq. ft.
Area of midship section	—	184 sq. ft.	184·6 sq. ft.	191 sq. ft.	168 sq. ft.	176 sq. ft.	105 sq. ft.	95 sq. ft.	105 sq. ft.	98 sq. ft.	126 sq. ft.	202 sq. ft.	54 sq. ft.
Displacement on trial	855 tons.	533 tons.	491 tons.	512 tons.	422 tons.	465 tons.	258 tons.	210 tons.	276 tons.	180 tons.	266 tons.	532 tons.	67 tons.
Co-effi. of displacement	—	·37	·38	—	·4	·42	·403	·393	·451	·41	·46	·38	·454
Displacement per inch immersion at L.W.L.	—	7·6 tons.	7·5 tons.	6·6 tons.	6·3 tons.	5·8 tons.	5 tons.	4 tons.	5 tons.	3·6 tons.	4·2 tons.	6·7 tons.	1·86 tons.
Midsection abaft centre length L.W.L.	—	4ft.	9ft. 7in.	3ft.	9ft.	5·5ft.	5·8ft.	12·5ft.	9ft.	7ft.	0·7ft.	8ft.	—
Centre of buoyancy do.	—	1ft.	5·2ft.	3·6ft.	0·7ft.	1·5ft.	4ft.	6ft.	3·96ft.	2·5ft.	1·7ft.	2·9ft.	2 5ft.
Area of wet surface ...	—	4880 sq. ft.	5100 sq. ft.	—	4500 sq. ft.	—	3212 sq. ft.	2800 sq. ft.	3136 sq. ft.	2190 sq. ft.	2755 sq. ft.	—	1084 sq ft.
Ballast	105 tons.	79 tons.	20 tons.	60 tons.	—	70 tons.	30 tons.	—	12 tons.	—	40 tons.	63 tons.	13 tons.
Builders' tonnage	720	664	529	507	451	—	—	280	268	182	240	565	85
Registered tons, gross and net	560 & 339	328 & 223	365 & 214	344 & 205	301 & 193	266 & 177	189 & 128	142 & 103	187 & 107	132 & 85	167 & 80	334 & 227	49 & 33
Fore-and-aft engine and boiler space......	—	26ft.	39ft. 6in.	31ft. 6in.	39ft.	28ft. 6in.	31ft.	28ft. 6in.	33ft. 3in.	—	27ft. 6in.	80ft.	16ft.
Diameter of cylinders.	25in. & 50in.	20in. & 40in.	23in. & 42in.	23in. & 44in.	23in. & 42in	21in. & 36in.	18in. & 36in.	16in. & 24in.	16in. & 32in.	24in. & 24in.	18in. & 32in.	24in. & 42in.	10in & 16½in
Stroke of pistons	30in.	24in.	30in.	24in.	30in.	24in.	22in.	24in.	24in.	18in.	24in	21in.	12in.
Revolutions	—	100	100	98	95	93	111	113	124	85	146	100	184
I.H.P. on trial...........	717	300	600	359	494	226	330	272	237	240	330	—	101
Condensing surface ...	—	—	1104 sq. ft.	1000 sq ft.	1104 sq. ft.	—	630 sq. ft.	—	480 sq. ft.	—	512 sq. ft.	—	390 sq. ft.
Vacuum....................	26·5in.	—	26·6in.	26in.	26in.	—	26in.	—	26in.	23in.	28in.	—	26¾in.
Diameter of boiler	—	—	13ft. 4in.	13ft.	12ft. 9in.	—	11ft. 3in.	—	10ft. 3in.	9ft. 3in.	10ft.	—	7ft.
Length of boiler	—	—	10ft. 3in.	9ft.	9ft. 9in.	—	10ft.	—	8ft 9in.	10ft.	8ft. 6in.	—	6ft. 6in.
Heating surface	2049 sq. ft.	—	1296 sq. ft.	1380 sq. ft.	1126 sq. ft.	—	1041 sq. ft.	—	725 sq. ft.	—	696 sq. ft.	—	333 sq. ft.
Grate surface	76·5 sq. ft.	—	55·6 sq. ft.	48 sq. ft.	49 sq. ft.	—	66 sq. ft.	—	36 sq. ft.	38 sq. ft.	30 sq. ft.	—	12·6 sq. ft.
Working pressure......	80lb.	60lb.	80lb.	75lb.	80lb.	70lb.	70lb.	60lb.	70lb.	45lb.	80lb	60lb.	80lb.
Coal-bunkercapacity...	100 tons.	80 tons.	100 tons.	80 tons.	70 tons.	70 tons.	50 tons.	30 tons.	60 tons.	50 tons.	33 tons.	80 tons.	8 tons.
Total weight of machinery with steam up	—	58 tons.	88 tons.	80 tons.	86 tons.	68 tons.	55 tons.	48 tons.	50·4 tons.	—	50 tons.	—	14 5 tons.
Diameter of propeller	—	—	10ft	10ft.	10ft.	—	9ft.	—	7ft. 9in.	7ft.	7ft.	—	4ft. 7¼in.
Pitch	16ft.	—	14·75ft.	13ft.	14·8ft.	11ft. 6in.	11ft.	—	10ft.	15ft.	10ft.	—	6ft. 3in.
Surface of all blades...	—	—	32·2ft.	—	—	—	16 sq. ft.	—	14·629ft.	—	13·5ft.	—	6 25 sq. ft
Number of blades.....	—	—	—	—	—	3	3	—	4	3	3	—	4
Wet surface per I.H.P.	—	16	8·5	—	9·1	—	10·7	12	13·22	11·6	8·34	—	10·7
Heating surface per I.H.P.	2·9 sq. ft.	—	2·16 sq. ft.	3·08 sq. ft	2·23 sq. ft.	—	3·47 sq. ft.	—	3·06 sq. ft.	—	2·11 sq. ft.	—	3·3 sq. ft.
Grate surface per I.H.P.	·105 sq. ft.	—	·092 sq. ft.	·133 sq ft.	0·1 sq. ft.	—	0·22 sq. ft.	—	·152 sq. ft.	—	0·091 sq. ft.	—	·12 sq. ft.
Speed on trial	12 knots.	8·9 knots.	13·02 knots.	10·2 knots.	12·5 knots.	9·34 knots.	11·7 knots.	10·52 knots.	10·147 knots.	11·75 knots.	11·2 knots.	10·2 knots.	9·1 knots.
Co-efficient for speed $\frac{S^3 \times D^{\frac{2}{3}}}{I.H.P.}$	220	155	228	188	221	214	186	150	186·4	216	185	—	122
Co-efficient for speed $\frac{S^3 \times M}{I.H.P.}$	—	432	—	—	672	633	510	524	462	661	567	—	406

risen from firemen or stokers. The Board of Trade qualifications for second-class engineer are as follows: "The candidate must be twenty-one years of age, and have served an apprenticeship to an engineer, and prove that during the period of his apprenticeship he has been employed in the making and repairing of engines; or if he has not served an apprenticeship, three years' employment in a factory or workshop where engines are made or repaired will suffice; but in either case he must also have served one year at sea in the engine-room. If he has not been apprenticed to an engineer, nor worked in a factory for the stipulated time, four years' service at sea in the engine-room will qualify him. However or wherever he may have been employed, he must be able to give a description of boilers and the methods of staying and tubing them, together with the use and management of the different valves, cocks, pipes, and connections. He must know how to correct defects from accidents and decay, and to repair them, and to understand the use of the barometer, thermometer, hydrometer, and salinometer. He must state, when asked, the causes, effects, and remedies for incrustation; and his educational attainments must be ample enough to pass in the first five rules of arithmetic and decimals, besides which he will be questioned as to the construction and fixing of paddle and screw engines."

In selecting a "second class engineer" to put in charge of a yacht's engines, one who is fresh from the factory or with only one year's sea service should be avoided; but the "donkey man," who has worked up to his certificate at sea will be a reliable engineer to engage.

The owner of a steam yacht who desires to become acquainted with the working of his engines should obtain the following books: "The Safe Use of Steam" (price 6*d.*), Crosby, Lockwood, and Co., 7, Stationers' Hall-court; "The Marine Steam Engine for the Use of the Officers of Her Majesty's Navy," by Richard Sennett, Longmans, Paternoster-row, 1882; and a "Manual of Marine Engineering," by A. E. Seaton, Griffin, and Co., Exeter-street, Strand. The latter has become a standard work.

CHAPTER XIII.

RULES AND FORMULÆ OF USE IN DETERMINING THE QUALITIES OF A YACHT.

All areas bounded by curved lines are treated by naval architects as parabolas of different orders, and their areas and centres of gravity are usually found by a computation of ordinates, the formulæ for which are generally known as "Simpson's Rules." An ordinate, as here understood, is any line projected at right angles to the base of a curve. In the diagram annexed (Fig. 139) A B is the base line of the curve A C B. The

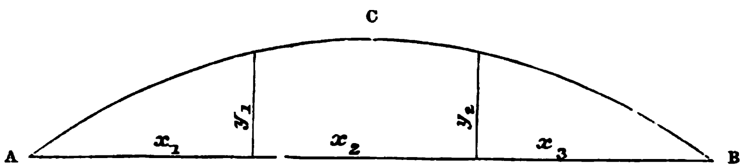


Fig. 139.

lines y_1 y_2 are ordinates. The intervals x_1 x_2 x_3 are called abscissæ; in this work they will be always referred to as "intervals," or as "longitudinal intervals."

The area inclosed in a curved line is expressed in algebraic symbols by $\int y\, d\, x$. $\int$ is the integral sign, being a kind of *s*, the initial of the Latin *summa*, sum (the integral is the sum of the differentials); *y* is the quantity summed; and *d* the differential of *x*.

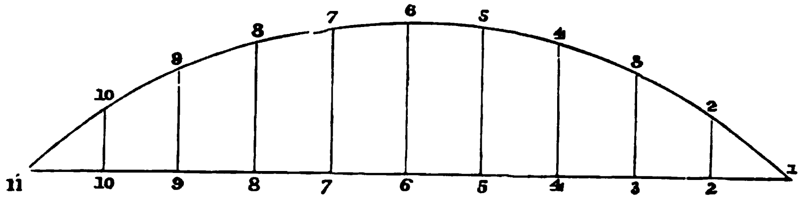


Fig. 140.

To find the area of a plane: Divide the base line into any even number of longitudinal intervals or spaces; but there must be an even number of spaces, so as to get an odd number of ordinates. From the

point marking each interval draw an ordinate at right angles to the base line to meet the curve; number the ordinates from left to right, or from right to left, as may seem most convenient (see Fig. 140).

RULE.—First take the lengths of even ordinates (commencing at No. 2 and ending at No. 10) and add the lengths together; multiply this sum by 4. Next take the odd ordinates (commencing at No. 3 and ending at No. 9) and add them together; multiply this sum by 2. Then add these two products together, with the sum of the first and the last ordinates (No. 1 and No. 11); multiply the total sum so gained by $\frac{1}{3}$ the common interval between the ordinates. The product will be the required area $\int y\ d\ x = \frac{x}{3}\ (y_1 + 4\ y_2 + 2\ y_3 + 4\ y_4 + 2\ y_5 + 4\ y_6 + 2\ y_7 + 4\ y_8 + 2\ y_9 + 4\ y_{10} + y_{11})$. This is called "Simpson's First Rule."

This is the rule in most common use among naval architects, and is capable of any degree of exactness by increasing the number of intervals, even though the curves are very sharp.

Simpson's second rule is more intricate than the first rule, and not so accurate; still it may be at times necessary to use the rule where an even number of intervals cannot be obtained without some inconvenience or trouble. The number of intervals must be any multiple of 3 (as $3 \times 4 = 12$), and thus the number of ordinates will be one in excess of the multiple 12—that is 13; the formula is $\int y\ d\ x = \frac{3}{8}\ x\ (y_1 + 3\ y_2\ y_3 + 2\ y_4 + 3\ y_5 + 3\ y_6 + 2\ y_7 + 3\ y_8 + 3\ y_9 + 2\ y_{10} + 3\ y_{11} + 3\ y_{12} + y_{13})$.

The ordinates 4, 7, 10 (multiples of every third interval), are termed dividing ordinates, and have to be multiplied by 2, or, in other words, the formula requires that the ordinate at every third interval must be so multiplied; the intermediate ordinates 2, 3, 5, 6, 8, 9, 11, 12, are multiplied by 3. To these products are added the sum of the two end ordinates, and the whole sum is multiplied by $\frac{3}{8}$ the common interval between the ordinates.

A modification of this rule applied to figures divided into four equal intervals is: Add together the two end ordinates, 3 times the sum of the two intermediate ordinates, and multiply by $\frac{3}{8}$ the common interval.

In computing the area of a quadrant of a circle 12ft. radius by ordinates at six equal intervals, the error was found to be nearly $\frac{1}{100}$ of the whole. This error can be reduced to $\frac{1}{250}$ by placing the ordinates one-half the distance nearer each other, that is, by making the number of equal intervals 12 instead of 6. The error can be further reduced to $\frac{1}{800}$ of the whole by making the number of intervals 24. The error, however, is so small where the curves are gradual, as they usually are in the water-lines of a ship, as to be scarcely appreciable.

It may not always be convenient to double the whole number of

ordinates in the curve; or it may be unnecessary to do so if there be a long "straight of breadth" in any part of the curve. In such cases half intervals may be used, as shown in Fig. 141.

It will be observed that extra ordinates have been put in at half intervals where the curve becomes oblique to the ordinates, and the ordinary multipliers (4 and 2) will be altered, as shown in the table, for

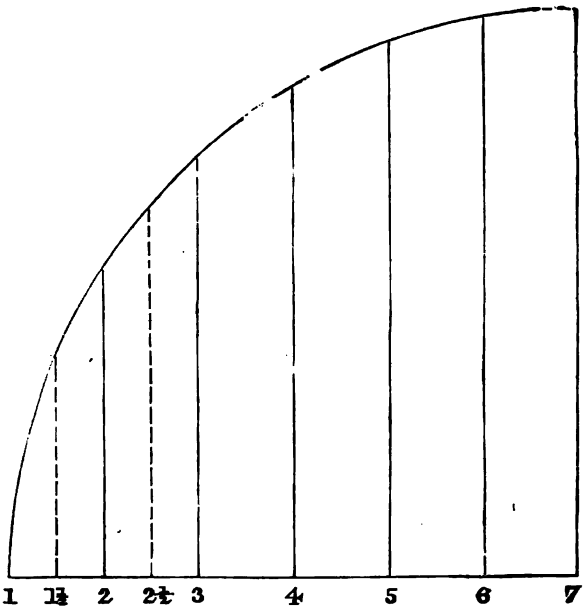


Fig. 141.

that part of the curve where the intervals are doubled. The ordinates are those of a quadrant of 12ft. radius, and are 2ft. apart.

Simpson's First Rule.

No. of ordinates.	Length of ordinates.	Simpson's multipliers.	Products to be summed.
1	0·0	½	0·0
1½	4·8	2	9·6
2	6·6	1	6·6
2½	7·9	2	15·8
3	8·9	1½	13·3
4	10·4	4	41·6
5	11·3	2	22·6
6	11·8	4	47·2
7	12·0	1	12·0

168·7
× ⅓ interval of 2ft. 2

3)337·4
112·46 = area in square feet.

Simpson's Second Rule.

No. of ordinates.	Length of ordinates.	Simpson's multipliers.	Products to be summed.
1	0·0	1	0·0
2	6·6	3	19·8
3	8·9	3	26·7
4	10·4	2	20·8
5	11·3	3	33·9
6	11·8	3	35·4
7	12·0	1	12·0

148·6
× ⅜ interval 2

297·2
3

8)891·6
111·45 = area.

With two places for decimals, the area given is 112·68 square feet. The exact area of a quadrant is known to be 113·1 square feet, so the small error here apparent can be considered of no importance in the general calculations a naval architect has to make. The area given by using Simpson's First Rule without half intervals is 112 square feet, and by the Second Rule 111·45 square feet; and, unless the curve be very full, as in the quadrant, and unless unusual accuracy be required, the calculator need scarcely ever trouble about putting in ordinates at half intervals.

A rule called the "Trapezoidal rule" can be used occasionally if great accuracy be not required, or if the curves be not abrupt. The rule is $\int y \, d x = x \left(\frac{1}{2} y_1 + y_2 + y_3 + y_4 + \frac{1}{2} y_5\right)$: that is, take half the sum of the two end ordinates y_1 and y_5 and add to the sum of all the other ordinates; multiply this sum by the common interval, and the product will

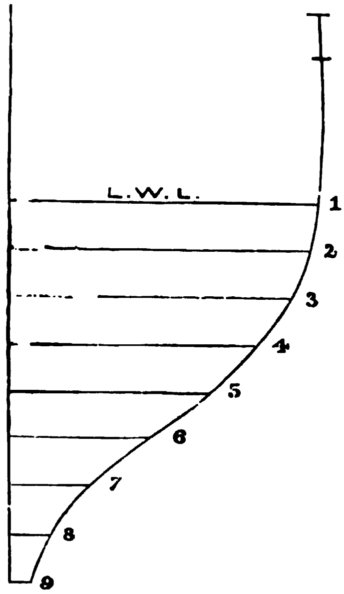


FIG. 142.

be the required area, approximately. Any number of intervals or ordinates can be used for this rule. Applied to a quadrant of 12ft. radius, the rule gives an area of 110 square feet—a variation from the true area much too considerable to be admitted into calculations—in short, the rule is never used except for "straight" curves, such as the lower water lines.

It has now been shown in what manner the areas of the horizontal sections (or water lines as they are usually termed) are calculated, and the areas of vertical "athwart ship" sections are calculated in a precisely similar way. Let Fig. 142 be any vertical transverse section of a ship such

as the midship section, then 1, 2, 3, 4, 5, 6, 7, 8, 9 are ordinates, and have to be summed in the way previously described by Simpson's rule.

The even ordinates 2, 4, 6, 8 lengths of the even ordinates measured from the curve to the vertical line are taken out, added together, and multiplied by 4. Next the odd ordinates 3, 5, 7 are taken out, added together, and multiplied by 2. The sum of these two products is then taken, and added to the sum will be the sums of the lengths of 1 and 9, the two "end ordinates in this case being the L.W.L." The whole sum has then to be multiplied by one-third the common interval between the ordinates, and the product will be the area required.

It is usual to make the ordinates of the water-lines that pass through the vertical sections shown on the body plan serve as the ordinates for the respective areas of the vertical sections, so that one set of ordinates will do for the whole operation of calculating the displacement and centre of buoyancy of a ship.

When a plane is bounded by a parabolic curve, a base line and two straight sides at right angles to the base (see Fig. 143), the area may be

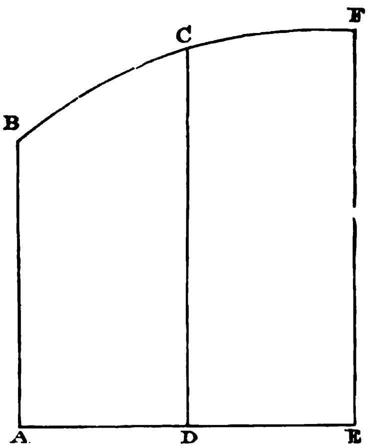


Fig. 143.

found thus: Add together the two ends ordinates A B and E F and four times the middle ordinate C D; multiply this sum by $\frac{1}{3}$ the interval between the ordinates, and the product will be the area. This rule is founded on the same reasoning as Simpson's "first rule."

To calculate the area contained in either space right or left of the ordinate C D, the rule will be to find the area of C F D E: Multiply the ordinate C F five times; the middle ordinate C D eight times; add these two products together, and from their sum subtract the other ordinate A B; multiply the remainder by $\frac{1}{12}$ the common interval, and the product will be the required area.

If the area A B C D is required: Multiply A B five times and C D eight times; sum the products and subtract E F; Multiply the remainder by $\frac{1}{12}$ the common interval, and the product will be the required area.

To find the area of a parabolic segment as shown (Fig. 144), multiply the base A B by $\frac{2}{3}$ the perpendicular height C D.

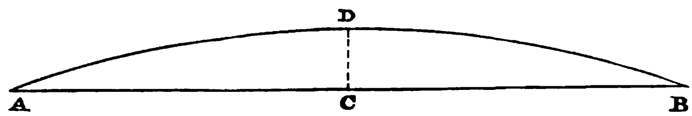


Fig. 144.

The area of a triangle is found by multiplying its base A B (Fig. 145) by half its perpendicular height, C *d*; C *d* is set off at right

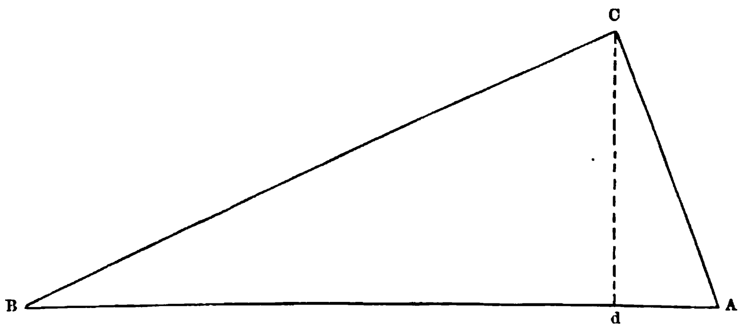


Fig. 145.

angles to A B, and is measured to the opposite angle C. The formula is $\frac{\text{base} \times \text{perpendicular}}{2}$.

The area of a trapezoid is thus found: Add together the top side

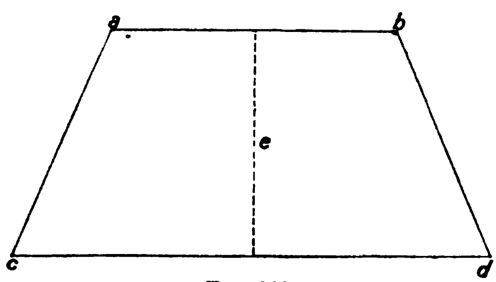


Fig. 146.

(Fig. 146) *a b* and the bottom side *c d*, and multiply this sum by the perpendicular *e*; divide by 2, and the quotient will be the area $\frac{ab + cd \times e}{2}$ = area.

To find the area of a plane contained within any angle, O P Q, and its arc P Q (Fig. 147): Divide the plane into any number of angular

intervals by the radii O P, O R, O S, O T, which are called polar co-ordinates, and radiate from the point O. Measure the length of each radius or ordinate, as O P, O R, &c.; take the half squares and put them into Simpson's First Rule; that is, add together the half squares of O P and O Q; four times the sum of the half squares O R, and O T; and twice the sum of the half square of S O; multiply the total sum by $\frac{1}{3}$ the angular interval expressed in circular measure, and the product will be the area required.

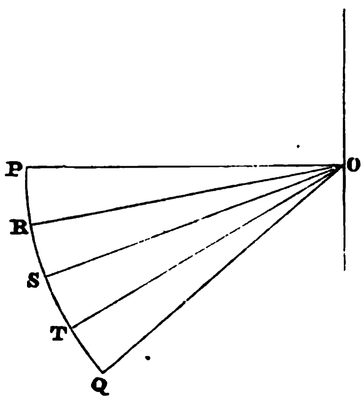


FIG. 147.

The algebraic expression for computing areas by polar co-ordinates is $\int \frac{r^2}{2} \, d\,\theta$, where r signifies any radius of the plane, and θ the angle made by any radius with either the outside radii.

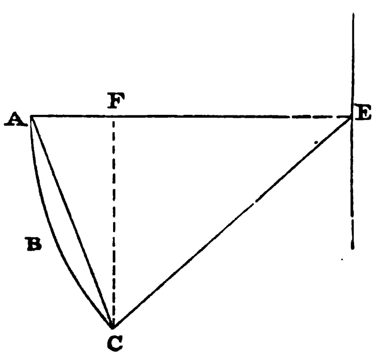


FIG. 148.

A table of factors for computing the length of circular arcs will be found in the Appendix. The half squares will be taken from any table of squares and cubes and halved.

A simpler way to find the area of such a shaped figure is to cut off the arc or segment A B C (Fig. 148); then find its area by rule 9 before given (Fig. 144); next find the area of the triangle, A C E, by rule 10.

F C will be the perpendicular for the triangle, and A E the base. Add the area of the parabolic segment of the area to the triangle for the required whole area.

The volumes of solids which are of such irregular shapes as ships are found by a further application of Simpson's rules. That is, each area is treated as an ordinate in the calculation. We will presume the solid to be of a shape similar to that depicted by Fig. 149. The areas of the ends of the vessel represented by 1 and 7 may be indefinitely small, according to the peculiarities of the vessel. Then 1, 2, 3, 4, 5, 6, 7 are areas of cross sections which represent planes in the calculations. The areas of these planes are found by the process already described; then each area is treated as an ordinate, and these "area-ordinates" are summed by one of Simpson's rules in the manner heretofore explained. Thus, the contents of the solid is given in cubic feet.

To ascertain the volume of any wedge-shaped solid, of which Fig. 147 may be considered a transverse section, conceive the wedge to be cut into angular layers in a longitudinal or fore-and-aft direction by the co-

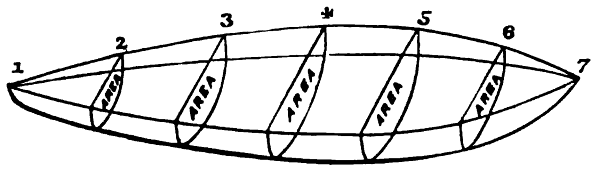


FIG. 149.

ordinates O P, O R, &c.; each of these layers will be subdivided by ordinates represented by the figures (refer to Fig. 150) 1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11. Take the half squares of the ordinates of each layer (in turn) and sum them according to Simpson's first rule; this will give the moment of each plane about the axis O. V. Take the "moments" so obtained and operate upon them as if they were co-ordinates, the summing of which, by Simpson's rule, will be the same as before illustrated (Fig. 147); that is, sum the moments of the planes O P V U, O R V W, O S V X, O T V Y, and O Q V Z, by Simpson's First Rule, and multiply the sum by $\frac{1}{3}$ the angular interval expressed in circular measure.

$$(\tfrac{1}{3}\theta \, OPVU + 4ORVW + 2OSVX + 4OTVY + OQVZ.)$$

Of course such a figure can be subdivided into any convenient number of angular intervals, but care must be taken to always select whichever of Simpson's rules suits the number of intervals chosen.

The volume of the wedge (Fig. 150) may also be found by computing the area of each transverse section of the wedge at the points 1, 2, 3, 4, &c., by polar co-ordinates (as shown in Fig. 147), and then summing

these transverse areas by one of Simpson's rules, each area so found being treated as an ordinate.

The volume of the wedge (Fig. 150) may be found in yet another way, by operating upon each angular transverse section 1, 2, 3, 4, &c., in the manner described in Fig. 148, calculating the area of the triangle and its parabolic segment. The areas must be summed in the usual way, according to Simpson's First Rule. This rule will be found most simple in practice if merely the volume of the wedge has to be calculated, and a detailed example of its application will be found further on.

The contents of an ordinary wedge is found by multiplying the area of its base by $\frac{1}{2}$ its perpendicular height. The contents of a frustrum of a wedge (*i.e.* a wedge with its thin end cut of) is found by the following formula: $A = \frac{p}{2}\ (b + c)$, where b area of the base, c area of top, p perpendicular; A solid contents of the wedge.

The solid contents of a cylindrical body, such as the mast, boom or

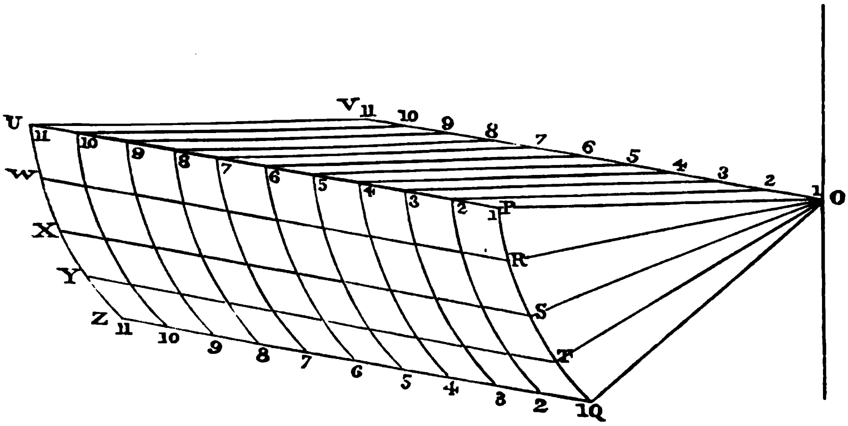


FIG. 150.

yard of a ship, is found by multiplying its length by the mean area of its cross section. The area of a section at any point would be found by multiplying the square of the diameter at the point by ·7854, which is the rule for finding the area of a circle. The mean cross area of a boom or other spar would be found by finding the area at three or more equidistant points. These areas would be added together, and their sum divided by the number of areas taken.

CENTRE OF GRAVITY.

The centre of gravity of such an area or plane as shown on page 321 (Fig. 140) is found by multiplying the ordinates by the number of their stations, from No. 1 station; the products or moments so found are then operated upon by one of Simpson's rules. The result so obtained is then

divided by the area of the plane, and the quotient multiplied by the distance the ordinates are apart; this product will be the distance the centre of gravity is from No. 1 station. This gives the centre of gravity of the figure in a longitudinal direction only; but inasmuch as the figure represents but one-half of the real figure, and as its corresponding half is of exactly similar form, the centre of gravity of the whole figure will lie in the line 1, 11 (Fig. 140).

The centre of gravity of a solid or of any portion of a solid immersed in water is found in a similar manner, the areas being treated as ordinates. Each area is multiplied by its number, from No. 1 station; the products are summed by one of Simpson's rules, and the result divided by the contents of the body expressed in cubic feet; the quotient multiplied by the distance between the sections will give the distance the centre of gravity is from No. 1 section.

The foregoing rules give the centre of gravity of a plane or body in a longitudinal direction; the position must also be found in the vertical direction. The rule to determine the position as to depth of the centre of gravity in such a body as the immersed portion of a ship is as follows: Take the areas of each of the longitudinal planes, such as Fig. 140 (which can be said to represent the water-line of a ship) and treat them as ordinates; proceed (in exactly the manner explained for the vertical sections) to operate upon these area ordinates by one of Simpson's rules, and divide the result by the contents of the solid expressed in cubic feet; multiply the quotient by the distance the horizontal or longitudinal planes are apart, and this product will be the distance the centre of gravity of the figure is from the initial plane, or below the plane of flotation.

The term "centre of gravity" here used, as applied to a body floating, is equivalent to centre of buoyancy—the more correct term is centre of gravity of displacement. It is, in fact, the centre of the hole made in the water by the body floating. It must be understood that "centre of gravity of displacement" and "centre of buoyancy" are convertible terms; the "centre of gravity" of the weight of a ship is always understood to mean the "centre of gravity" of the collective weights of a ship, and is altogether distinct from centre of buoyancy or centre of gravity of displacement. The mode for ascertaining the "centre of gravity" of a ship is explained farther on.

It will be sometimes necessary to find the common centre of gravity of two bodies or planes which in some way exercise a joint force.

The formula will be $x = \frac{d \times w}{W + w}$, where W weight of one body, w weight of the other body, d the distance the centres of gravity of the two bodies are apart, x the distance the common centre of gravity is from the centre

of gravity of W. Then multiply the weight of w by the distance d, its centre of gravity, is from the centre of gravity of W, and divide by the combined weight of the two bodies. The quotient will be the distance the common centre of gravity of the two bodies is from the centre of gravity of W—represented by the line x in the diagram (Fig. 151).

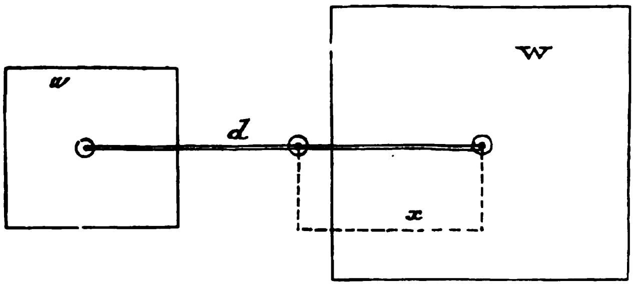


FIG. 151.

(In the case of planes, the areas of the respective planes will be treated as the weight in the foregoing illustration. See also Fig. 153, page 332.)

If it is required to know how far the centre of gravity of a body will be shifted by the shifting of any portion of its weight, the rule will be found in the following expression: $x = \frac{W \times d}{s}$, where W the weight to be shifted, d the distance the weight is moved, s the total weight of the body, and x the distance the centre of gravity is moved. Supposing two tons of ballast be taken out of yacht's hull and placed on her keel 3ft. lower down, the whole weight of the yacht being 20 tons.

3·0 feet.
2·0 tons.
20 tons)6·0(0·3 = feet.
6·0

That is, the centre of gravity of the yacht would be brought 0·3 feet lower, or $3\frac{5}{8}$ inches.

If it is required to know how far a weight must be moved to shift the centre of gravity a certain distance, then the formula is $d = \frac{S \times x}{W}$; that is, multiply the total weight of the body (S) by the distance the centre of gravity has to be moved (x), and divide by the weight W to be shifted.

20
·3
2)6·0
3·0 feet.

That is, the weight would have to be shifted 3ft.

The centre of gravity of a triangle is found by bisecting (*i.e.*, dividing the middle of the line A B in a; from a produce a line to C; two-thirds

of this line set off from C towards *a* will give the position of the centre of gravity of the triangle *d* (Fig. 152).

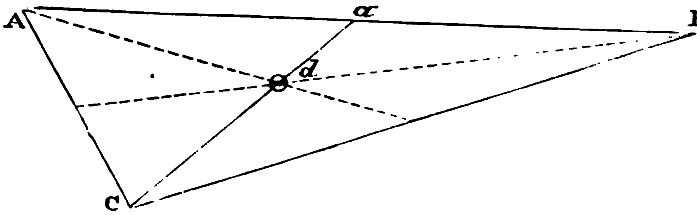


FIG. 152.

Or each side of the triangle can be bisected, and a line produced to an opposite angle, as shown by the ticked lines on Fig. 152. The point, *d*, where the lines intersect will be the centre of gravity of the figure. This, in practice, will be found the more accurate way.

To find the centre of gravity of a trapezium (Fig. 153):

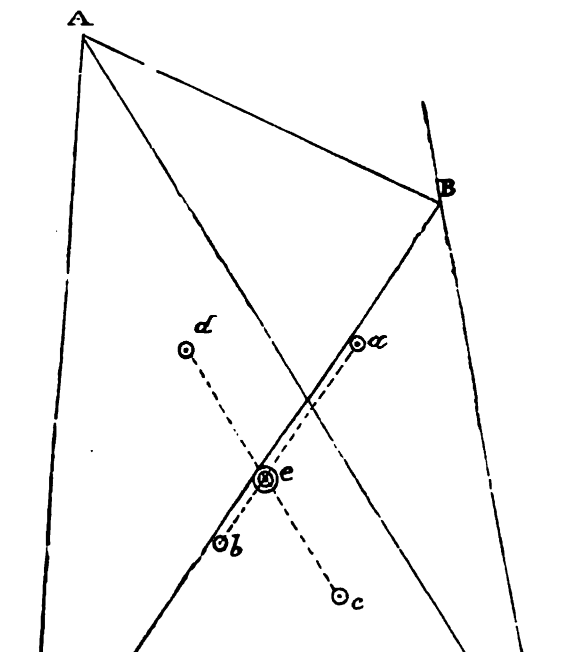


FIG. 153.

Divide the figure into four triangles, as shown by A B D, B C D, A B C, and A D C, and proceed to find the centre of gravity of each

triangle according to the rule given in Fig. 152; then draw a line from the centre of each triangle to the centre of its corresponding triangle; the point of intersection will be the centre of gravity of the figure; let A B C be one triangle, with its centre of gravity at *a*; let A D C be its corresponding triangle, with its centre of gravity at *b*; then

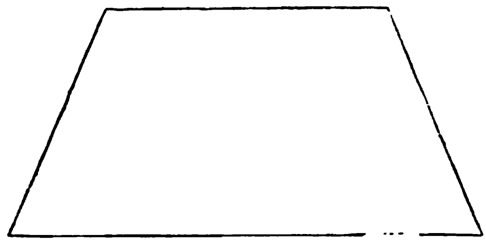


FIG. 154.

join *a b*. Next let B C D be a triangle with its centre of gravity at *c*; let its corresponding triangle be A B D, with its centre of gravity at *d*; join *c d*, and the point where *c d* and *a b* intersect will be the centre of gravity of the figure *e*.

The centre of gravity of a trapezoid (Fig. 154) can be found in a precisely similar way.

To find the centre of gravity of a figure by experiment: Suspend the object (Fig. 155), and let fall a plumb-line from the point of suspension;

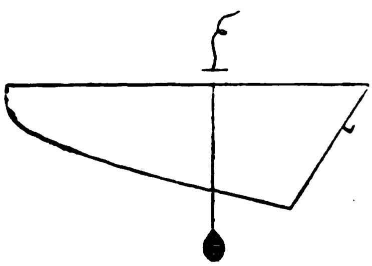


FIG. 155.

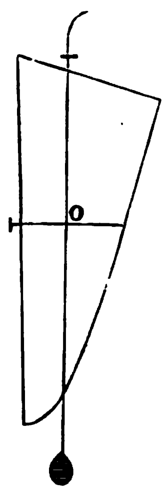


FIG. 156.

mark where the plumb-line cuts the figure; then suspend the figure from another point (Fig. 156), and let fall a plumb-line as before. The point where the plumb-lines intersect will be the centre of gravity of the figure O (Fig. 156).

To ascertain the mean breadth of a plane, divide the area by length of the base, and the quotient will be the required mean breadth.

To find the mean area of the sections of a solid, divide the volume (expressed in cubic feet) by the length of the solid, and the quotient will be the required mean area.

Similarly, the mean area of the water-lines can be found by dividing the contents of the solid by its depth.

If the displacement has been calculated without the plank (termed "moulded displacement," because it is to the moulding or shape of the frames only) the actual displacement can be found by multiplying the area of the immersed surface in square feet by the thickness of the plank, the latter, of course, in decimals of a foot. Or the moulded displacement will be to the true displacement as the cube of the breadth of the figure (at the L.W.L.) moulded is to the cube of the breadth (at the L.W.L.) with the plank on: or as $a : b :: c : d$ and $d = \frac{b^3 \times c}{a^3}$.

Where a the moulded breadth; b the breadth with the plank on; c the moulded displacement; d the displacement with the plank on.

Also the displacement of vessels of similar form can be found by this equation; thus, say the length of a yacht is 40ft. and her displacement 15 tons; and the length of another yacht on the same lines, but to a different scale, so as to be 50ft. on the water-line, then their displacement to each other will be $\frac{50^3 \times 15}{40^3} = 29{\cdot}3$ tons; and similarly the areas of figures are to each other as the squares of either of their leading dimensions.

Say the area of wetted surface of a yacht 50ft. long is 340 sq. ft., then the area of wetted surface of a yacht 40ft. long, built on the same lines (but to a different scale) will be $\frac{40^2 \times 340}{50^2} = 217{\cdot}6$ sq. ft.

The displacement may also be found from a model (see pages 2 and 3).

The usual method, however, of finding the displacement of vessels of similar form but of different dimensions is by the cube of the scale of comparison, as explained on page 3, the following notation being observed:

A the dimensions of one vessel, say, the breadth or length.

B the same dimensions in another vessel.

D the displacement of one vessel.

Da the displacement of another vessel, then

$Da = \left(\frac{B}{A}\right)^3 \times D$. This, of course, assumes that the B dimensions are larger than the A. If the opposite is the case, then the formula will take this form: $Da = \frac{D}{\left(\frac{A}{B}\right)^3}$.

Areas of surfaces or of similar planes will be found by using the square of the scale of comparison instead of the cube.

CHAPTER XIV.

CALCULATING THE DISPLACEMENT, CENTRE OF BUOYANCY, ETC.

THE examples of the calculations which are here given relate to the cutter yacht Kriemhilda, the lines of which are depicted in Plate X.

It is generally found most convenient to number the ordinates from the stem, and this plan has been followed in the example about to be given. There is, however, nothing arbitrary in the matter, and so far as the accuracy of the result of the calculations is concerned, the ordinates might just as well be numbered from the sternpost.

The ordinates must be numbered as shown on the drawing (Plate X.), No. 1 commencing at the fore side of the stem; the last ordinate is No. 17, which passes through a portion of the rudder. It must be noted that the ordinates represented by ticked lines at half intervals, and numbered 1½, 2½, 15½, and 16½ in the Sheer Plan, do not refer to any operation in calculating the displacement. The use of these ordinates will appear further on.

The common custom is to divide the length of the load water-line into an equal number of parts, beginning at the fore side of the stem and terminating at the after side of the stern post. This will always be the most convenient rule when the stern post is upright; but it may be sometimes more convenient (to get the distance between the ordinates to represent the spacing of the frames) to produce the last ordinate through the rudder, as shown in the example about to be given. By such a process a portion of the rudder, is, of course, included in the calculations, and the remaining piece must be included in the final calculation of the whole volume; the piece of the rudder, marked P in Plate X., which remains abaft the ordinate, must be calculated separately. The Kriemhilda, for instance, is 79·5ft. in length on the L.W.L. from the fore side of stem to aft side of stern post, and this length subdivided into sixteen equal parts would give a rather awkward length of interval to set off accurately;

consequently a portion of the rudder was treated as a part of the load water-line, to render the length of the latter equal to 80ft., which admitted of subdivisions of 5ft.

Generally, however, the sections are put in at the intervals where the frames will come, as otherwise a second drawing might be required for the mould loft. In such a case it may be possible that the first or last section may not be a whole interval from the stem or stern post. The calculation can be made from the section, or an imaginary piece can be added beyond the stem or stern post to complete the interval; but this must be allowed for both in the displacement and in setting off the calculated distance the centre of buoyancy is from the stem.

It will be found in the drawing of Kriemhilda that the stem, or fore foot, is rounded away so that at No 2 water-line the space between Nos. 1 and 2 ordinates does not quite equal 5ft., the common interval. If the water-line at this end were very full it would be necessary to make a fresh set of ordinates for No. 2 water-line, and, indeed, for all the water-lines, to ensure accuracy; but, inasmuch as the lines at the fore end of the vessel (as is usually the case in yachts) are very straight, this need not be done. It can be assumed that all the water-lines are extended to meet No. 1 ordinate or perpendicular dropped from the stem at the L.W.L., and so the subdivisions made for the load water-line will answer for each succeeding water-line. Allowance can afterwards be made for the small excess introduced. When No. 6 water-line is arrived at, it will be best to make a change in numbering the ordinates; that is, take No. 2 as No. 1, No. 3 for No. 2, and so on all through, making No. 16 No. 15.

It is usual to calculate the displacement in halves, as the half sections of the vessel only are given. But formerly naval architects practised another subdivision, that of calculating the displacement of the fore-body and after-body separately; there seems to have been no weighty reason for this latter subdivision, and as the practice involves additional labour without any advantage, it is now seldom carried out, except for some exceptional purpose.

It is occasionally the practice to leave a portion of the vessel which comes under the description of "keel" out of the calculation for the displacement; when this is done a supplementary calculation has to be made, an example of which is given. There is no reason why the water-lines should not be continued to the under side of the keel if found convenient to do so; if this be done the necessity for the supplementary calculation will not exist. To ensure accuracy in the calculations, the

water-lines should never be further apart than $\frac{1}{6}$ the depth taken amidships from the load-line to the rabbet-line.

It should be stated that the lengths of the ordinates—in fact all the measurements—were taken from a drawing made to a $\frac{1}{2}$in. scale, and not from the reduced drawing shown on Plate X. Of course the larger the scale the drawing is made to, the more nearly will it accord to the shape of the ship, and the smaller will be the errors likely to be made in taking off measurements owing to the unfairness of the drawing, or from want of precision in using the scale.

It will be seen that some licence is permissible even in calculations of this nature, and it has already been pointed out how convenience in fixing intervals for ordinates can still further be studied. A drawing might be made with No. 1 station placed at 1ft. more or less abaft the fore side of the stem or the sheer drawing, and the other stations at 6ft. or some other distance apart. In such a case the section or ordinate at No. 1 station can be used as No. 1 area or ordinate for the calculation, the outlying portion being afterwards calculated separately. When this happens the operator in calculating the positions of the various centres of gravity must be careful to remember that all linear measurements must be taken from No. 1 station, and not from the fore side of the stem.

In the calculations for the Kriemhilda two places for decimals have been retained throughout; for general work one place is sufficient, "giving and taking." However, in some cases even three and four places for decimals must be used, as, for instance, in the final operation of division and multiplication to determine the position of the centre of gravity of displacement.

It might here be noticed that the practice among naval architects is to have paper ruled to receive all the quantities that come into the operation of calculating a ship's displacement, tabulated—ordinates, multipliers, areas, and all—so that read one way the sums of the ordinates, and areas, &c., of the cross sections are found, and read the other way those of the water-lines are found. No doubt an expert professional man may save himself some labour in making figures by such a method; and the method of using the displacement sheet will be explained farther on.

The scales used for taking off the lengths of the ordinates are divided decimally into tenths and not duodecimally into twelfths. It will often be found convenient to measure the lengths of the ordinates, or half breadths, by a "half scale;" that is, if the drawing is made to half an inch scale, measure all the half breadths or ordinates by a quarter of an inch

scale. However, the distances or intervals between the ordinates must be measured by the *whole* scale. The result will, of course, be the *whole* displacement instead of half the displacement.

In the examples of the calculations now given for the water-lines of the Kriemhilda the first column of figures merely represents the numerical distinction of the ordinates; the second column represents the lengths of the ordinates measured by a scale from the middle line of the half-breadth plan to the point where the ordinates cut the curve of the particular water-line whose area is being calculated. Under the head of "even ordinates" will be found those ordinates whose stations from No. 1 are even numbers, as 2, 4, 6, &c.; the "odd ordinates" are those whose stations are odd numbers, as 3, 5, 7, &c. It will be seen that the respective lengths of these even and odd ordinates are added together, and the two sums are multiplied by 4 and 2 respectively, in accordance with Simpson's First Rule. The further operation is sufficiently explained in the tables given.

It will be noted that when No. 6 water-line is arrived at the change alluded to above is made in the numbering of the ordinates, and a further change is made at No. 7 water-line. This change is made because —owing to the rounding up of the fore foot and the rounding away of the rudder—the lower water-lines are much shortened. In the case of No. 6 water-line the ordinate numbered 1 at the stern and the ordinate numbered 17 at the rudder are dropped altogether. The other ordinates remain exactly the same, excepting that they are renumbered—No. 2 ordinate of the L.W.L. becomes the No. 1 of the sixth water-line, No. 3 ordinate is No. 2, and so on. In the case of No. 7 water-line a further ordinate is dropped at the stem, but none at the stern; the ordinates have to be renumbered, and, as their total will be fourteen, the Trapezoidal Rule, instead of Simpson's First Rule, will be used. In general practice it will not be found necessary to renumber the ordinates, as the small volume added by assuming all the water-lines to be of the same length as the L.W.L. can easily be estimated when the calculation is completed.

In the calculations which follow the method of determining the centre of gravity of the load water plane is given. There is but little object, however, in knowing its position, although it has been the practice of some designers to find the centre, and arrange the centres of succeeding water-lines below, according to some rule. The rule would be of value as a means of comparison if no other method existed, but comparisons as to form can be made by easier and more reliable methods, as we have already shown.

Area and Centre of Gravity of Load Water Plane or Line of Flotation.

It will be seen that the lengths of the ordinates are put into a column opposite the numbers or station 1, 2, 3, 4, &c. Those which come opposite the even numbers, such as 2, 4, 6, 8, &c., are added together and multiplied by 4, and those which come opposite the odd numbers of 3, 5, 7, 9, &c., are multiplied by 2; the two products are then added together (with the sum of the end ordinates) and multiplied by $\frac{1}{3}$ (one third) the interval; that is, by 5ft. the interval, and then divided by 3.

Distinguishing Nos. of Ordinates.	Lengths of Ordinates in linear feet.
1	0·20
2	1·35
3	2·75
4	4·20
5	5·60
6	6·75
7	7·65
8	8·30
9	8·60
10	8·70
11	8·60
12	8·28
13	7·90
14	7·05
15	5·35
16	2·70
17	0·30

Even Ordinates.		Odd Ordinates.	
2	1·35	3	2·75
4	4·20	5	5·60
6	6·75	7	7·65
8	8·30	9	8·60
10	8.70	11	8·60
12	8·28	13	7·90
14	7·05	15	5·35
16	2·70		
Sum	47·33 × 4	Sum	46·45 × 2
	4		2
	189·32		92·90

Sum of even ordinates × 4 =	189·32
Sum of odd ordinates × 2 =	92·90
Sum of two end ordinates......	·50
	282·72
× $\frac{1}{3}$ 5ft., distance the ordinates are apart	5
	3)1413·60
	471·20 = half area of load water plane in square feet.

$$\text{Mean half breadth of ordinates} = \frac{471{\cdot}20}{80} = 5{\cdot}89\text{ft.}$$

But a better plan than this is to multiply the ordinates by 4 and 2 as they stand in their respective columns, but care must be taken that 4 is placed opposite the even numbers and 2 opposite the odd numbers.

The cubic contents of the keels of modern yachts must be calculated separately.

The centre of gravity is found as follows: Another column is added to the right, as shown (page 340), for the multipliers 1, 2, 3, 4, 5, 6, &c., always recollecting that 0 comes opposite No. 1. The "first products," are multiplied by 1, 2, 3, 4, 5, 6, &c., and placed in another column to the right

termed moments. These moments are added together and divided by the total sum of the "first products." In this case $\frac{2430\cdot72}{282\cdot72} = 8\cdot597$, the quotient 8·597 is multiplied by the longitudinal interval, and the product is the distance the centre of gravity is from No. 1 station.

Nos. of Ordinates.	Lengths of Ordinates.	Simpson's Multipliers.	First Products.	Multipliers.	Column for Moments.
1	0·20	1	0·20	0	0·00
2	1·35	4	5·40	1	5·40
3	2·75	2	5·50	2	11·00
4	4·20	4	16·80	3	50·40
5	5·60	2	11·20	4	44·80
6	6·75	4	27·00	5	135·00
7	7·65	2	15·30	6	91·80
8	8·30	4	33·20	7	232·40
9	8·60	2	17 20	8	137·60
10	8·70	4	34·80	9	313·20
11	8·60	2	17·20	10	172·00
12	8·28	4	33·12	11	364·32
13	7·90	2	15·80	12	189·60
14	7·05	4	28·20	13	366·60
15	5·35	2	10·70	14	149·80
16	2·70	4	10·80	15	162·00
17	0·30	1	0·30	16	4·80

```
 282·72 ............ )2430·72(8·597
      5              2261·76
3)1413·60             168·960
  471·20              141·360
                       27·6000
                       25·4448
                        2·15520
                        1·97904
                         ·17616
```

8·597 multiplied by the interval (5ft.) between the sections gives the distance the centre of gravity of the plane is from No. 1 section.

```
 8·597 × 5ft. long. inter.
     5
42·985ft.
```

42·985ft. = distance the centre of gravity the L.W.L. is aft No. 1 section (fore side of the stem in this instance).

Displacement by Horizontal Sections or Water-lines.

The half area of the load water plane has been already calculated, and the half areas of the other water-lines, or water sections, or planes, as they are more correctly termed, must now be proceeded with. The ordinates are measured and set out in a column just as the ordinates of the load water plane were; but, as the centre of gravity of the planes below the L.W.L. will not be required, the calculation will be more simple as follows:

WATER-LINE No. 2.

Distinguishing Nos. Of Ordinates.	Lengths of Ordinates in linear feet.		
1	0·00	1	0·00
2	1·10	4	4·40
3	2·17	2	4·34
4	3·45	4	13·80
5	4·70	2	9·40
6	5·90	4	23·60
7	6·90	2	13·80
8	7·70	4	30·80
9	8·10	2	16·20
10	8·20	4	32·80
11	8·10	2	16·20
12	7·55	4	30·20
13	6·66	2	13·32
14	5·05	4	20·20
15	2·85	2	5·70
16	0·90	4	3·60
17	0·30	1	0·30
			238·66
			5
			3)1193·30
			397·76 = half area.

WATER-LINE No. 3.

Distinguishing Nos. Of Ordinates.	Lengths of Ordinates in linear feet.
1	0·00
2	0·80
3	1·63
4	2·60
5	3·55
6	4·57
7	5·60
8	6·45
9	6·95
10	7·00
11	6·60
12	5·75
13	4·50
14	2·75
14	1·15
16	0·55
17	0·30

303·566 = half area of water-line No. 3 as calculated in the manner No. 2 was.

WATER-LINE No. 4.

Distinguishing Nos. of Ordinates.	Lengths of Ordinates in linear feet.
1	0·00
2	0·55
3	1·10
4	1·75
5	2·40
6	3·10
7	3·80
8	4·50
9	4·90
10	4·90
11	4·35
12	3·40
13	2·20
14	1·20
15	0·70
16	0·50
17	0·30

198·00 = half area of water-line No. 4.

WATER-LINE No. 5.

Distinguishing Nos. of Ordinates.	Lengths of Ordinates in linear feet.
1	0·00
2	0·40
3	0·75
4	1·10
5	1·50
6	1·90
7	2·20
8	2·60
9	2·75
10	2·70
11	2·40
12	1·70
13	1·10
14	0·70
15	0·50
16	0·40
17	0·30

114·50 = half area of water-line No. 5 similarly calculated.

Water-line No. 6.

In this water plane only fifteen ordinates are computed, Nos. 1 and 17 having been dropped. It will be seen that this (No. 6) water-line commences at nearly the No. 2 ordinate of the other water-lines, and is made to end at the No. 16 ordinate; so the reason for making the change is obvious. It will be observed that the ordinates have to be renumbered, No. 2 becoming No. 1, No. 3 No. 2, and so on.

Distinguishing Nos. of Ordinates on the plan.		Lengths of Ordinates in linear feet.
2	1	0·20
3	2	0·45
4	3	0·60
5	4	0·95
6	5	1·10
7	6	1·30
8	7	1·40
9	8	1·50
10	9	1·20
11	10	0·90
12	11	0·70
13	12	0·55
14	13	0·50
15	14	0·40
16	15	0·30

59·16 = half area of water-line No 6.

By the Trapezoidal rule the area of this water-line was found as under:

Sum of intermediate ordinates.........	11·15
Half sum of two end ordinates	·25
	11·80 × 5ft. interval.
	5
	59·00 = area in square feet.

Water-line No. 7.

No. 1 ordinate of this water-line is the No. 3 of the load water-line. The area is computed by the Trapezoidal rule.

Distinguishing Nos. of Ordinates on the plan.	Lengths of Ordinates in linear feet.	
1	0·25	
2	0·30	
3	0·40	
4	0·50	
5	0·60	
6	0·60	
7	0·70	
8	0·60	
9	0·60	
10	0·50	
11	0·40	
12	0·40	
13	0·40	
14	0·00	0·60

Sum of intermediate ordinates.........	6·00
Half sum of two end ordinates.........	0·12
	6·12 × 5ft.
	30·26 = half area of water-line. No. 7.

Water-line No. 8.

This water-line commences at No. 6 and terminates at No. 16 in the Sheer Plan. The ordinates will be renumbered as follows:

Distinguishing Nos. of Ordinates on the plan.	Length of Ordinates.	
6	0·0	
7	0·2	
8	0·2	
9	0·3	
10	0·3	
11	0·3	
12	0·3	
13	0·2	
14	0·2	
15	0·2	
16	0·0	21·5
		5
		10·5 = Half area No. 8.

Water-line No. 9.

This water-line will be the underside of the keel at No. 13 section where there is a straight piece equal to 3ft. The half-thickness is 0·3ft., and the area can be put down at 1 sq. ft. $0·3 \times 3 = 0·9$.

The Displacement and Centre of Buoyancy.

Having found the half areas of the various water-lines, the next step in the calculations will determine the displacement, and the calculator will find that this operation exactly resembles the process he has just put the water-lines through to compute their respective half areas.

The half areas of the water-lines, as they have just been found, must be taken and put into a table as if they were the ordinates of a new curve; these areas will be in this instance summed by Simpson's First Rule, and multiplied by $\frac{1}{3}$ the common interval between the water-lines. In this instance the interval is 1ft. 6in., or 1·5ft.

To find the centre of buoyancy, the several half areas of the water-lines will be put into a column next the products and multiplied by the number of their stations below the load water-line, as shown in the example about to be given, and the sum of the moments thus arrived at will be divided by the sum of the products; and that quotient multiplied by the distance the water-lines are apart will give the depth the centre of gravity

of the displacement (centre of buoyancy) is below the load water-line or plane of flotation.

Nos. of Water-lines.	Half Areas of Water-lines.	Simpson's Multipliers.	Products.	Multipliers for centre of gravity.	Moments.
1	471·20	1	471·20	0	1591·04
2	397·76	4	1591·04	1	1215·88
3	303·97	2	607·94	2	2376·00
4	198·00	4	792·00	3	916·00
5	114·50	2	229·00	4	1183·20
6	59·16	4	236·64	5	367·20
7	30·60	2	61·20	6	294·00
8	10·50	4	42·00	7	8·00
9	1·00	1	1·00	8	

4033·02 ....)7951·32(1·972
1·5ft. = distance water-lines are apart × ⅓ 1·5 | 4032·02 | 1·5 = interval.
2016·010 | 3919·300 | ·9860
4032·02 | 3628·818 | 1·972
3)6048·030 | 290·4820 | 2·9580 = distance the centre of buoyancy is below the L.W.L.
7)2016·010 cubic feet. | 282·2414
5)288·00 | 8·24060
| 4·06404
57·6 = half displacement in tons. | ·17656
2
115·2 = whole displacement.

[NOTE.—The cubic feet are divided by 7 and 5, because 35 is a multiple of 7 and 5, and there are 35 cubic feet of salt water in a ton.]

Thus far we make the displacement of the Kriemhilda to be 115·2 tons, but there is still a small portion of the rudder to consider. The upper portion of the rudder (P) which was not considered in the calculations can be safely disregarded, as it will very nearly balance the outlying piece that was assumed to be added to the fore foot during the process of calculating the areas of the first five water-lines. The piece of rudder, however, between Nos. 16 and 17 sections, and Nos. 5 and 7 water-lines, must be calculated. For this purpose refer to the rule illustrated by Fig. 143, page 325.

The dividing ordinate A measures 4ft., B 5ft., C 2ft.:

A ...... 4ft. × 4 | B ...... 5·
4 | C ...... 2·
16 | Sum of two end ordinates...... = 7·
Sum of dividing ordinate × 4 = 16·
23· × ⅓ interval of 1·5ft.
1·5
11·5
23·
The mean thickness of the piece is 0·3ft., then 11·5 | 3)34·5
·3
3·45 | 11·5 = area

That is, 3·45, or say 3·5 cubic feet, or 0·1 ton, as $\frac{3\cdot5}{35} = 0\cdot1$ to be added to the displacement, making the total 115·3 tons.

If the displacement had been calculated without the keel, say to No. 7 water-line, there would be still the keel to calculate. Take the scale and measure the water-line, No. 7, from the fore foot to the back of the rudder where the latter is cut by the water-line; this length will be found to equal 68ft.; divide this length into four equal intervals of 17ft. each, and number them as shown on the dotted lines, y_1, y_2, y_3, y_4, y_5. Treat these dotted lines as ordinates, and sum them according to Simpson's First Rule. As the ends at the fore foot and rudder go off to a "feather" edge, no measurements can be put down for 1 and 5 ordinates.

Nos. of Ordinates.	Lengths of Ordinates.
1	0·0
2	1·5
3	2·5
4	3·0
5	0·0

```
 1·5          2·5
 3·0            2
 ----         ---
 4·5          5·0
   4
 ----
 18·0
  5·0
 ----
 23·0 × ⅓ 17ft.
  17·
 ----
 161·0
 230·
 -----
3)391·0
 -----
 130·3 = area
```

The mean thickness of the dead wood is 0·5ft., then 130·3
·5
65·15 cubic feet = 1·8 ton.

That is, 65·15 cubic feet, or 1·8 tons, have to be added to the displacement, to which add 4·6ft. for the piece of rudder.

```
65·15                                      7)68·65
 3·5                                       5)9·80
68·65 cubic feet to be reduced to tons.     1·98 tons.
```

That is, 1·98 tons have to be added to the displacement of the vessel to No. 7 water-line, found to be 113·25 tons.

```
113·25
  1·98
------
115·23 = total displacement of the Kriemhilda.
```

If the under side of the keel forms a perfectly straight line from the stern post to the fore foot (where the latter is intersected by the lower water-line), the keel portion of the vessel can be treated as a wedge, and its volume found by multiplying the area of the base (or thick end at the stern) by half the perpendicular height or length. Thus, treating the under side of Kriemhilda's keel as "straight," we have the following sum. For area of base of wedge—

Depth of keel at stern.........	3ft.
Mean width ditto	0·6
	1·8 = area.
½ 68ft. length =	34
	7·2
	54
	61·2 cubic feet.

The error by this rule for Kriemhilda will thus be under 4 cubic feet —no serious matter; still, when there is no great difference in the amount of labour, it will always be best to use those rules which yield a minimum of error.

In ordinary cases it would be unnecessary to regard the slight difference the buoyancy represented by the volume of the keel would make on the position of the centre of buoyancy; but in the case of modern yachts with fin or bulb keels, the centre of gravity of the whole mass should be found by the example shown on page 131, and given in detail on page 352.

The centre of gravity of the keel must be first found. To do this, treat its vertical fore and aft section as a parabola, with the seventh water-line as its base. The centre of gravity of a parabola is $\frac{2}{5}$ of its perpendicular from the base. Presuming the under side of the keel to be straight, its centre of gravity could be readily found in the manner the centre of gravity of a triangle is determined. However, in this case we must take the parabolic form, and the calculation will be sufficiently correct for practical purposes.

The perpendicular of the parabola is 3ft., $\frac{2}{5}$ of which = 3
2
5)6
1·2ft.

That is, the centre of gravity of the keel is 1·2ft. below the seventh water-line; its distance below the load water-line will therefore be 10·2ft., seeing that the seventh water-line is 9ft. below.

Multiply the distance the centre of gravity of the keel is from the centre of buoyancy by the contents of the keel, and divide by the whole displacement. (C.B., for the displacement to the No. 7 water-line = 2·83ft. below L.W.L.)

7·36 = distance centre of gravity of keel is below C.B.
1·8 = contents of keel in tons.
5·888
7·36

Displacement 115·06)13·248(0·11
11·506
1·7420
1·1506
·5914

Add this quotient (0·11) to 2·83
0·11
2·94 = distance the centre of gravity of displacement is below the L.W.L.

DISPLACEMENT PER INCH OF IMMERSION.*

It will always be necessary to know what the displacement of a vessel is per inch of immersion at or near the load water-line; that is, occasions may arise when it will be important to ascertain how many inches a vessel will be lightened or immersed by removing or adding any given weight. This can only be ascertained by knowing the vessel's displacement per inch of immersion at the L.W.L. The rule is a very simple one: Take the total area of the L.W.L.—equal 942·4 square feet in Kriemhilda—divide by 12 and 35. Thus:

```
                12)942·4
7 × 5 = 35.      7)78·53
                 5)11·22
                   2·24 tons = displacement of Kriemhilda per inch of immersion.
                                        at L.W.L.
```

Having found the displacement of the Kriemhilda per inch of immersion, it is easy to ascertain how much she would be further immersed on coming from salt into fresh water. Kriemhilda's displacement = 115·06 tons; her displacement per inch of immersion = 2·24 tons. The weight of fresh water is about $\frac{1}{40}$ less than that of salt water.

```
   115·06           2·24        89·60)115·06(1·28in.
 ----------           40              89·60
 2·24 × 40          -----             ------
                    89·60             25·460
                                      17·920
                                      ------
                                       7·5400
                                       7·1680
                                       ------
                                        ·3720
```

That is, the Kriemhilda would float a little more than 1¼in. deeper in fresh, or river, water than she would in salt. Taking clean fresh water and clean salt water the displacement in cubic feet is $\frac{1}{40}$ (one fortieth) greater in fresh water that in salt.

* The following rough rule will give the displacement per inch of immersion, and be accurate enough for ascertaining the weight of ballast to remove to lighten the yacht one inch, or to put in to deepen her: Multiply the length on the load-line by the breadth on the load-line and divide the product by 600. $\left(\frac{L \times B.}{600}\right)$ The quotient will be the weight in tons or fractions of a ton. This rule would hold good for the load-line if the yacht were lightened more than three or four inches or deepened to that extent. The rule is based on the assumption that the area of the load water-line or any water-line is 0·7 of the circumscribing parallelogram formed by length and breadth. That is to say, the length and breadth multiplied together and again multiplied by 0·7 will (approximately) give the area of the water-line. Divide this product by 12, and the area is reduced to cubic feet, and divide again by 35 and the quotient will give tons or fractions of a ton. By this rough rule the displacement per inch at any part of the hull of the vessel (if the measurements are taken at the part) can be found approximately $\left(\frac{L \times B \times \cdot 7}{12 \times 35}\right) = \left(\frac{L \times B.}{600}\right)$

The rule can be illustrated by the case of Kriemhilda = 79·5ft. on L.W.L., and 17·2ft. breadth on L.W.L.

$$\frac{79{\cdot}5 \times 17{\cdot}2}{600} = 2{\cdot}279 \text{ tons per inch of immersion at the L.W.L.}$$

Displacement by Vertical Transverse Sections and the Longitudinal Position of the Centre of Buoyancy.

In the preceding calculations it was shown how the centre of gravity of displacement is determined with regard to its depth below the L.W.L.; but ordinarily it is more important to know its position in a longitudinal, or fore and aft, direction, as the relative fulness or fineness of the fore-body and after-body can be estimated if the longitudinal position of the centre of buoyancy be known, also the position to place the lead keel (see page 8).

To determine the longitudinal position of the centre of buoyancy, the displacement must be calculated over again by vertical transverse sections instead of horizontal sections or water-lines, and this will further serve as a check upon the first calculation.

In selecting the ordinates from the tables used for the water-lines to use for the transverse vertical sections, care must be taken to note if the ordinates underwent any change in numbering, as the sixth and seventh water-lines did in this example; but as a rule it will be better to take the measurements from the drawing, as this will serve as a check.

No. 1 section, it will be seen, is the fore side of the stem, and, unless the stem be an upright one, no area can be put down for No. 1 section. If the stem be vertical, the area of its half-siding must be computed as a section.

It will be noted that the distance these ordinates are apart is necessarily the distance the water-lines are apart.

Section 2.

Nos. of Ordinates.	Lengths of Ordinates.	Simpson's Multipliers.	Products.
1	1·35	1	1·35
2	1·10	4	4·40
3	0·80	2	1·60
4	0·55	4	2·20
5	0·40	2	0·80
6	0·30	4	1·20
7	0·00	1	0·00

11·55 × ⅓ distance the ordinates (water-lines) are apart.
1·5

5·775
11·55

3)17·325

5·775 = half area in square feet.

SECTION 3.

Nos. of Ordinates.	Lengths of Ordinates.	Simpson's Multipliers.	Products.
1	2·75	1	2·75
2	2·17	4	8·68
3	1·63	2	3·26
4	1·10	4	4·40
5	0·75	2	1·50
6	0·45	4	1·80
7	0·25	1	0·25

22·64 × ½ interval.
1·5

11·320
22·64

3)33·960

11·32 = half area.

SECTION 4.

Nos. of Ordinates.	Lengths of Ordinates.	Simpson's Multipliers.	Products.
1	4·20	1	4·20
2	3·45	4	13·80
3	2·60	2	5·20
4	1·75	4	7·00
5	1·10	2	2·20
6	0·60	4	2·40
7	0·35	1	0·35

35·15 × ½ interval.
1·5

17·575
35·15

3)52·725

17·57
Keel 0·13

17·70 = half area.

Upon reference to the Sheer Plan, it will be found that the keel at this (No. 4 section) extends below the No. 7 water-line; therefore this piece of the keel to its mean half siding (half breadth) must be calculated and added to the section (and the same for the other sections—5, 6, 7, &c.). The mean half siding is 0·2ft., and the depth 0·66ft., and ·2 × ·66 =.13ft Therefore the half area of the section will be—

Area to No. 7 W.L. = 17·57
Piece of keel........ ... 0·13

17·70 sq. ft.*

* In modern yachts like Isolde, Audrey, Stephanie, &c., the same method will have to be followed.

Section 5.

Nos. of Ordinates.	Lengths of Ordinates.
1	5·60
2	4·70
3	3.55
4	2·40
5	1·50
6	0·95
7	0·45

24·175
Keel... 0·225

24·400 = half area.

Section 6.

Nos. of Ordinates.	Lengths of Ordinates.
1	6·75
2	5·90
3	4·57
4	3·10
5	1·90
6	1·10
7	0·50

30·295
Keel... 0·455

30·750 = half area.

Section 7.

Nos. of Ordinates.	Lengths of Ordinates.
1	7·65
2	6·90
3	5·60
4	3·80
5	2·20
6	1·30
7	0·60

35·925
Keel... 0·475

36·400 = half area.

Section 8.

Nos. of Ordinates.	Lengths of Ordinates.
1	8·30
2	7·70
3	6·45
4	4·50
5	2·60
6	1·40
7	0·65

40·725
Keel... 0·475

41·200 = half area.

Section 9.

Nos. of Ordinates.	Lengths of Ordinates.
1	8·60
2	8·10
3	6·95
4	4·90
5	2·75
6	1·50
7	0·70

43·35
Keel... 0·65

44·00 = half area.

Section 10.

Nos. of Ordinates.	Lengths of Ordinates.
1	8·70
2	8.20
3	7·00
4	4·90
5	2·70
6	1·20
7	0·65

42·975
Keel... 0·725

43·700 = half area.

Section 11.

Nos. of Ordinates.	Lengths of Ordinates.
1	8·60
2	8·10
3	6·60
4	4·35
5	2·40
6	0·70
7	0·60

39·90
Keel... 1·1

40·1 = half area.

Section 12.

Nos. of Ordinates.	Lengths of Ordinates.
1	8·28
2	7·55
3	5·75
4	3·40
5	1·70
6	0·70
7	0·50

35·14
Keel... 0·86

36·00 = half area.

Section 13.

Nos. of Ordinates.	Lengths of Ordinates.
1	7·90
2	6·66
3	4·50
4	2·20
5	1·10
6	0·55
7	0·40

28·57
Keel... 0·73

29·30 = half area.

SECTION 14.

Nos. of Ordinates.	Lengths of Ordinates.
1	7·05
2	5·05
3	2·75
4	1·20
5	0·70
6	0·50
7	0·40

20·675
Keel 0·625

21·300 = half area.

SECTION 15.

Nos. of Ordinates.	Lengths of Ordinates.
1	5·35
2	2·85
3	1·15
4	0·70
5	0·50
6	0·40
7	0·40

12·425
Keel 0·575

13·000 = half area.

SECTION 16.

Nos. of Ordinates.	Lengths of Ordinates.
1	2·70
2	0·90
3	0·55
4	0·50
5	0·40
6	0·30
7	0·00

5·70
Keel 0·30

6·00 = half area.

SECTION 17.

Nos. of Ordinates.	Lengths of Ordinates.
1	0·30
2	0·30
3	0·30
4	0·30
5	0·30
6	0·30

2·250 = half area.

The half areas of the various transverse or cross sections having been found, it remains to tabulate them for summation by Simpson's First Rule, as were the half areas of the different water sections.

The tables, it will be seen, have been arranged according to the method previously adopted.

Nos. of Vertical Sections.	Half areas of Vertical Sections.	Simpson's Multipliers.	Products.	Multipliers.	Moments.
1	0·00	1	0·00	0	00·00
2	5·77	4	23·08	1	23·08
3	11·32	2	22·64	2	45·28
4	17·70	4	70·80	3	212·40
5	24·40	2	48·80	4	195·20
6	30·75	4	123·00	5	615·00
7	36·40	2	72·80	6	436·80
8	41·30	4	165.20	7	1156·40
9	44·00	2	88·00	8	704·00
10	43·70	4	174·80	9	1573·20
11	41·00	2	82·00	10	820·00
12	36·00	4	144·00	11	1584·00
13	29·30	2	58·60	12	703·20
14	21·30	4	85·20	13	1107·90
15	13·00	2	26·00	14	364·00
16	6·00	4	24·00	15	360·00
17	2·25	1	2·25	16	36·00

1211·17...............)9935·56(8·2
× ⅓ 5ft. 5 9689·36 5

3)6055·85 246·200 41·0 = the distance the centre of buoyancy is from the stem.

7)2018·62 = cubic ft. 242·234

5)288·39 3·966

57·68
2

115·36 tons displacement.

This result agrees closely with the volume of the displacement found by water-lines.

[Assuming that the displacement by vertical sections had only been calculated to a plane, say at No. 7 water-line, the longitudinal position of the centre of gravity of the keel would have to be ascertained. To accomplish this the keel would be divided into a convenient number of vertical sections, and the centre of gravity found in the manner that the centre of buoyancy has been. The centre of gravity of the keel was found to be 37·68ft. abaft No. 3 section, and the centre of buoyancy 40·88ft. abaft fore side of stem; thus the centre of the keel would be 6·8ft. abaft the centre of buoyancy. The alteration to the previously ascertained position of the centre of buoyancy would be found by the following equation. The distance will be 6·8ft. Multiply by the volume of the keel expressed in tons, and divide by the whole displacement expressed in tons, thus:

$$
\begin{array}{rl}
 & 1{\cdot}8 = \text{contents of keel in tons.} \\
 & \underline{6{\cdot}8} = \text{distance its C.G. is from C.B.} \\
 & 1{\cdot}44 \\
 & \underline{10{\cdot}8} \\
\div \text{ displacement} = 115{\cdot}36) & 12{\cdot}240(0{\cdot}1\text{ft.} \\
 & \underline{11{\cdot}536} \\
 & \underline{0{\cdot}704}
\end{array}
$$

That is, the centre of gravity of displacement would be 0·1ft. further aft than found by the first calculation; or, the centre of gravity of displacement of the Kriemhilda is 40·98ft. abaft the fore side of the stem instead of 40·88ft.

By the previous calculation, which included the keel in the sections, it was found to be 41ft. abaft the fore side of stem. *However, when practicable the vertical transverse sections should always include the keel.*]

Co-efficient of Fineness.

The co-efficient of fineness is the proportion of the actual bulk of a body to what the bulk would be if the figure were a rectangular solid or parallelopipedon of the same extreme length, breadth, and depth; or the ratio of the actual area of a plane to the area of the parallelogram found by its extreme length and breadth.

The co-efficient of displacement is thus found:

$$\text{Co-efficient} = \frac{\text{Displacement in cubic feet.}}{\text{Length} \times \text{breadth} \times \text{mean draught.}}$$

The co-efficient of a plane with a curved outline will be found from:

$$\text{Co-efficient} = \frac{\text{Area in square feet.}}{\text{Length} \times \text{Breadth.}}$$

The Displacement Sheet.

As already explained, it is usual to rule paper to tabulate the ordinates on what is termed a "displacement sheet," so that read one way they represent the horizontal sections, and the other the ordinates of the vertical sections. When large series of calculations have to be made, time is saved by this method; but, in order to make the subject as clear as possible, the calculations of the horizontal and vertical sections have been separated in the foregoing examples, and those who only require occasionally to make such calculations will find it more convenient to follow these methods, ruling a small memorandum book in much the same way as found in the text.

The displacement sheet is ruled as shown in the example now given. The lengths of the ordinates are entered in the table in thick black figures (or say in red ink in practice), and are taken from the half breadth plan of Kriemhilda (Plate X.), but it will be noted that every other ordinate has been omitted, so that the intervals are 8 instead of 16. Also only one place for decimals has been kept. In noting the result of the calculations it will be found to be almost identical with the more elaborate operation previously described.

The ordinates read transversely make the vertical sections. "Simpson's multipliers" will be found in the top line above. As each ordinate is multiplied the product is entered under the black figures (in practice these products could be entered in blue ink).

After all the ordinates have been multiplied the products are added together and entered in blue ink in a column to the right, headed "Products for displacement" These last products are all added together, and this sum (1215·5) is multiplied by $\frac{1}{3}$ the longitudinal interval, 10ft. in this case (that is, multiplied by 10 and divide by 3). Next, multiply by $\frac{1}{3}$ the vertical interval or space between the water-lines, 1·5ft. in this case. (Multiply by 1·5 and divide by 3). The result is the $\frac{1}{2}$ displacement in cubic feet, which, multiplied by 35 (or, $7 \times 5 = 35$), gives the $\frac{1}{2}$ displacement in tons.

The fore and aft position of the centre of buoyancy is found, as shown by multiplying the "Products for displacement" by numerals, which operation gives the moments set out in the last column to the right. This operation is exactly similar to that described previously for calculating the centre of gravity.

The sum of the moments is 4996·8, which is divided by the sum of the "products for displacements" 1215·5. The quotient 4·1 is multiplied

by 10 the longitudinal interval, and the product is the distance in feet the C.B. is from the foreside of stem, or, to speak more correctly, from No. 1 station.

Simpson's multipliers for the water-lines are in the vertical column —second on the left of the sheet. As the ordinates are multiplied the products are entered by themselves in the spaces on the right in black ink; and when added together the sums are entered at the foot of the vertical column in the line "Products of ordinates of water-lines."

These products are in turn operated upon by another set of Simpson's multipliers and entered in the spaces below. They are subsequently taken out and summed in the manner shown in the left-hand lower corner.

The multipliers for the position of C.B. below the load water-line are entered in the horizontal row below the "products for areas," and are in turn summed as shown in the left-hand lower corner. The sum is 2427·6, which is divided by the sum of the areas 1215·5. The quotient 19·9 is multiplied by the space between the water-line = 1·5ft., and the product 2·985ft. is the distance the C.B. is below the L.W.L.

The area of the load water-plane and area of midship section, &c., are found as shown. Usually also, the displacement scale is calculated on the sheet from the areas of the water-lines. Sometimes, also, the metacentre is calculated, the cubes of the ordinates of the load water-line being entered in a column to the extreme right, and operated upon as shown farther on. However, the sheet as shown contains all the usual calculations.

Curve of Displacement or Curve of Areas of Sections.

It is usual to show the "form" of the displacement by a curve of sectional areas. This is a very simple operation. Project a base-line, A B, Fig. 157, to scale, to represent the length of load-line, which might include the portion of rudder projecting beyond the aft end of the load-line. On this line set up ordinates at the proper stations for the vertical sections. Next take the half (or whole) area of each vertical section and divide it by some convenient number. In the case of Kriemhilda. the half sections were taken from the table on page 351, and divided by 5 Thus the area of No. 2 section in 5·77 sq. ft., which, divided by 5, gives 1·15ft., which is set off on its proper ordinate to the same scale as the line A B. When all the sections have been so treated and set off on

their respective ordinates, a batten is taken, and the curve drawn through the points as shown in Fig. 157.

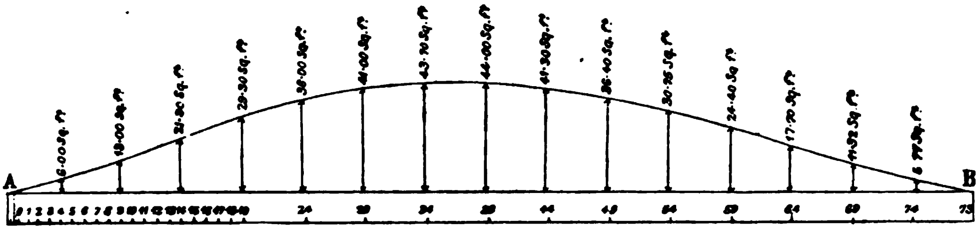


FIG. 157.

Sometimes a wave curve is drawn, and the areas of the various sections in a design are made to accord with the readings taken off the ordinates by the same scale (*ante* the "Wave Form" theory).

SCALE OF VERTICAL DISPLACEMENT.

The scale of displacement is intended to show the displacement of a vessel to any given line of flotation. The curved line A (Fig. 158)

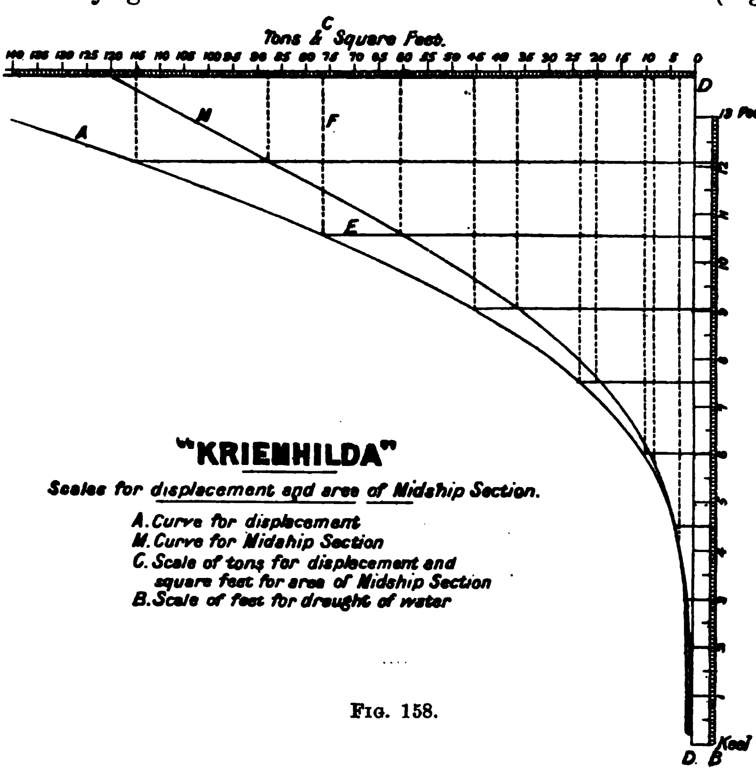


FIG. 158.

represents the growth of the displacement from the keel upwards; the vertical scale B represents the draught of water; the horizontal scale C

represents tons. To find the displacement to any given draught, draw an ordinate, E, at right angles to the perpendicular, D, from a point opposite the required draught; from the point where the line E intersects the curve A, draw a line, F, at right angles to E and parallel to D; the point where this line meets the scale of tons will show the displacement in tons.

On Fig. 158 the horizontal lines shown on the scale represent the water-lines, and are 1·5ft. apart.

To construct the curve A the displacement must be calculated to a series of water-lines, and there are several rules for doing this. The most obvious plan is to calculate the displacement to each water-line by the rules already explained for calculating the displacement by horizontal sections or water-lines. But in most cases it will be found that the simplest plan is to use a combination of rules.

For instance, to find the displacement to No. 2 water-line of Kriemhilda—that is, to the water-line 1ft. 2in. below the load water-line—it is not necessary to calculate the displacement to that line from the keel upwards, as by the rule explained (page 354) the volume contained in the space between No. 2 water-line and the load line can be calculated; having found this volume, it will be subtracted from the whole displacement, and thus the displacement to No. 2 water-line will be arrived at.

The rule will be: Multiply the area of the L.W.L. five times; multiply the area of the second W.L. eight times; add these two products together, and from their sum subtract the area of the third W.L.: multiply the remainder by $\frac{1}{12}$ the interval between the water-lines, and the product will be the required volume contained in the space between No. 2 W.L., and the L.W.L.

For the Kriemhilda we have the following quantities: (It must be noted that only the half areas of the water-lines are here taken.)

$$
\begin{array}{lr}
\text{Half area of L.W.L.} = 471{\cdot}2 \text{ sq. ft.} \times 5. & \\
 & 5 \\
 & \overline{2356} \\
\text{Half area of No. 2 W.L.} \times 8 = & 397{\cdot}76 \times 8. \\
 & 8 \\
 & \overline{3182{\cdot}08} \\
+\text{Half area L.W.L.} = 471{\cdot}2 \times 5 = & 2356{\cdot} \\
 & \overline{5538{\cdot}08} \\
-\text{Half area of third W.L.} \ldots\ldots\ldots & 303{\cdot}57 \\
 & \overline{5234{\cdot}51} \times \tfrac{1}{12}\ 1{\cdot}5\text{ft. interval.} \\
 & 1{\cdot}5 \\
 & \overline{2617{\cdot}255} \\
 & 5234{\cdot}51 \\
 & 12)\overline{7851{\cdot}765} \\
 & \overline{654{\cdot}313}
\end{array}
$$

This quantity = 654·31 cubic feet must be reduced to tons by the divisor 35 ($7 \times 5 = 35$).

$$
\begin{array}{l}
7\underline{)654\cdot31} \div 35. \\
5\underline{)93\cdot47} \\
\quad 18\cdot69 = \tfrac{1}{2} \text{ volume in tons, which} + 2. \\
\quad \underline{\quad 2} \\
\quad \underline{37\cdot38}
\end{array}
$$

That is, the volume of the layer between No. 2 and the load water-line equals 37·38 tons; this quantity subtracted from the whole displacement gives the displacement to No. 2 water-line thus:

$$
\begin{array}{l}
115\cdot25 \text{ tons} = \text{whole displacement.} \\
\underline{\ 37\cdot38} \\
\ \underline{77\cdot87} \text{ tons} = \text{displacement to No. 2 water-line.}
\end{array}
$$

Measure the draught (always taken from the lowest point of the keel) on the scale of feet to the second water-line; from this point draw the ordinate (*see* E, Fig. 158) which will represent the second W.L.; then find out 77·87 tons on the scale of tons, and let fall a perpendicular (*see* F, Fig. 158) from the point; the intersection of the ordinate E and perpendicular will give a point in the curve.

The displacement to No. 3 water-line must be found by Simpson's First Rule, thus (for the purpose of this calculation No. 3 is treated as No. 1 water-line):

	Half Area.	
No. 3 W.L.	305·57 × 1	= 305·57
4 "	198·00 × 4	= 792·00
5 "	114·50 × 2	= 229·00
6 "	59·16 × 4	= 236·64
7 "	30·60 × 1	= 30·60

$$
\begin{array}{l}
1593\cdot81 \times \tfrac{1}{2} \text{ interval } 1\cdot5\text{ft.} \\
\quad \underline{1\cdot5} \\
\ 796\cdot905 \\
\underline{1593\cdot81\ } \\
3\underline{)2390\cdot715} \\
7\underline{)796\cdot905} = \text{cubic feet.} \\
5\underline{)113\cdot843} \\
\quad 22\cdot768 = \tfrac{1}{2} \text{ volume} \times 2. \\
\quad \underline{\qquad 2} \\
\quad \underline{45\cdot536} \text{ tons.}
\end{array}
$$

Thus the displacement to the third water-line equals 45·536 tons. The draught to the third water-line must be found as before, and a perpendicular let fall from the scale of tons at the point 45·36 tons. The intersection of the ordinate drawn from the scale of feet to the perpendicular from the scale of tons will give another point in the curve.

The displacement to the fourth water-line must be found by an application of Simpson's Second Rule (*see* page 323), tabulating the half areas of the water-lines as before, commencing with No. 4 as No. 1.

No. 4 W.L.	....	198·00 × 1 = 198·00
5 „	....	114·50 × 3 = 343·50
6 „	....	59·16 × 3 = 177·48
7 „	....	30·60 × 1 = 30·60

749·58 × ⅜ interval of 1·5ft.
1·5

374·790
749·58

1124·370
3

8)3373·110

7)421·638 = cubic feet.

5)60·234

12·046 = ½ volume × 2.
2

24·092 tons.

The displacement to No. 4 water-line thus equals 24·092 tons, and to get a point for the curve the calculator must proceed as before.

To arrive at the displacement to No. 5 water-line a rule founded on Simpson's First Rule must be taken.

No. 5 W.L.	....	114·50 × 1 = 114·50
6 „	....	59·16 × 4 = 236·64
7 „	....	30·60 × 1 = 30·60

381·74 × ⅓ interval of 1·5ft.
1·5

190·870
381·74

3)572·610

7)190·870 = cubic feet.

5)27·267

5·453 = ½ volume × 2.
2

10·906

The displacement to the fifth water-line equals 10·906 tons, and the point for the curve must be found as before described.

The displacement to the sixth water-line will have to be found by the rule used to calculate the displacement to the second line. To five times the area of the fifth water-line add eight times the area of the sixth water-line; from this sum subtract the area of the seventh water-line, and multiply by $\frac{1}{12}$ the interval between the water-lines. The product will be the volume contained in the slice between the fifth and sixth

water-lines. Subtract this quantity from the displacement to the fifth water-line, and the remainder will be the displacement to the sixth water-line.

Fifth W.L. = 114·50		Sixth W.L. = 59·16
5		8
572·50		473·28
473·28		
1045·78		
Seventh W.L. 30·60		
1015·18 × $\frac{1}{12}$ interval 1·5ft.		
1·5		
507·590		
1015·18		
12)1522·770		
7)126·897 + 35.		
5)18·128		
3·625 = ½ volume × 2.		
2		
7·250 tons.		
Subtract 7·250 from 10·906		
7·250		
3·056		

Thus we find that the displacement to No. 6 water-line is 3·656 tons and this quantity must be used to get a point in the curve.

No further calculation is necessary, as the displacement to No. 7 water-line is already known by the calculation for the keel. The curve will radiate from the perpendicular D at a point representing the under side of the keel, and a batten bent to the "points" formed by the intersection of the ordinates which were projected from the scale of feet, and the perpendiculars dropped from the scale of tons, will enable the calculator to complete the operation by sweeping in the curve.

Curve of Immersion of Midship Section.

Upon reference to the half breadth plan of Kriemhilda it will be seen that a slightly larger section could be obtained between No. 9 and No. 10 ordinates than either of the sections produced in the body plan from the half breadths at No. 9 and No. 10 sections. This would be the midship section, and its area must be calculated. In the representation of the Kriemhilda's body plan (Fig. 159) farther on, the midship section is drawn and marked in the usual way by the symbol ⊗, and is situated midway between No. 9 and No. 10 sections. In the case of the Kriemhilda the midship section is not considered in the calculations for displacement and centre of

buoyancy, but it may be necessary to know the area of this the greatest transverse section.

It may so happen that when the position of the midship section has been determined that its station will come exactly right for one of the equal divisions of the length of the load water-line. If it does not it will be of no consequence for the purpose of the calculations; but for convenience, and to save time and trouble, the drawing for the calculations is generally so managed as to serve for the building draught as well.

Having found the area of the midship section, a curve showing its area for different draughts of water will be made; this operation will be analogous to the one for making the curve of displacement, and will be performed by a combination of rules. Upon reference to Fig. 158 it will be noted that the scale, C, of tons used for the displacement is also used for the area of midship section, "square feet" being substituted for "tons." The curve M represents the growth of area of the midship section from the keel upwards. The scale of feet for draught is B, the same as used for the displacement. The scale C is one of square feet for area.

The points for the curve are obtained by the intersection of the lines projected from the scale of square feet, and from the scale of linear feet for draught.

AREA OF MIDSHIP SECTION.

Nos. of Ordinates.	Lengths of Ordinates.	Simpson's Multipliers.	Products.
L.W.L. 1	8·75	1	8·75
2	8·23	4	32·92
3	7·10	2	14·20
4	5·00	4	20·00
5	2·82	2	5·64
6	1·40	4	5·60
7	0·60	2	1·20
8	0·35	4	1·40
9	·00	1	0·00
			89·71
$\frac{1}{3}$ × 1·5ft. interval			1·5
			44·855
			89·71
			3)134·565
			44·855 = half area in square feet.
			2
			89·710 whole area in square feet.

To make the curve for midship section (*see* M, Fig. 158) the calculator must proceed as for the displacement, using the ordinates of the section instead of areas of water-lines.

Calculating the Area of Immersed Surface.

The lengths of girths of each section in the body plan will be first measured by a pair of "dividers" opened to represent, say one foot, on the scale of the drawing. Or the frames can be measured by a spline batten marked to the scale or afterwards applied to the scale. The measurements will then be operated upon as shown in the example for Kriemhilda. Next the length of the curvature of the water-lines is measured by "dividers" in order to obtain the *mean* distance the vertical sections are apart. The distances the sections are apart are increased in an irregular manner if measured in a direction with the curvature of each water-line.

Each section or frame delineated on the body plan of the calculation drawing is measured from the underside of the keel to the load water-line. To the length of each frame is added the half siding of the keel.

The mean distance which the frames are apart (measured in line with the curves of the water-lines) will be thus found: Begin with the load water-line on the half breadth plan and measure the length of that line round its curve, beginning at No. 1 station (see Plate X.) at the stem and ending at No. 17 through the rudder. The length of this line will prove to be 82·5ft., and as there are sixteen intervals or spaces between the frames the required mean distance will be found $\frac{82\cdot5}{16} = 5\cdot156$ft. Thus the intervals between the frames on the curves of the load water-line will be 5·156ft. instead of 5ft. The required mean distance of interval for the other water-lines will be found in a similar manner; but the water-lines *below* No. 2 water-line should be measured from No. 2 station on the half-breadth plan, as the lower water-lines do not extend so far forward as the load water-line. For a similar reason No. 7 and No. 6 water-lines will be made to terminate at No. 16 station on the half-breadth plan. The *number* of intervals will, therefore, in those instances, be less than 16.

In calculating the area of wetted surface of a vessel it will be best, if she has a very much rockered keel, or fin, or fin bulb keel, to strike a *base line* (the top of the keel would do), and work from that for the lengths of the frames. The area of the keel or fin, &c., would then be calculated separately.

WATER LINES.

No. of Water-lines.	Mean Distances the Ordinates are apart.	Simpson's Multipliers.	Products.
1	5·156	1	5·156
2	5·142	4	20·568
3	5·133	2	10·266
4	5·065	4	20·260
5	5·022	2	10·044
6	5·001	4	20·004
7	5·000	1	5·000

÷ 3 the mean of Simpson's multipliers ....3)91·298

÷ number of intervals.........6)30·432

Mean distance the frames are apart = ... 5·072ft.*

FRAMES.

Nos. of Frames.	Half Girths of Frames.	Simpson's Multipliers.	Products.
1	0·0	1	0·0
2	8·4	4	33·6
3	9·8	2	19·6
4	11·0	4	44·0
5	11·8	2	23·6
6	12·9	4	51·6
7	14·0	2	28·0
8	14·8	4	59·2
9	15·4	2	30·8
10	15·6	4	62·4
11	15·7	2	31·4
12	15·8	4	63·2
13	15·4	2	30·8
14	15·4	4	61·6
15	15·2	2	30·4
16	12·0	4	48·0
17	6·5	1	6·5

÷ 3 mean of Simpson's multipliers .... 3)624·7

× 5·072 = mean distance that the frames are apart......... 208·2

5·072

·4164

14·574

1041·0

× 2............................. 1055·9904

2

Whole area of wetted surface = 1111·9808 sq.ft.

If the mean distance had been taken as 5ft. instead of 5·072ft., the actual mean distance, the result given would have been 2082 square feet, showing a difference of 30 square feet.

* Instead of measuring the distance between the frames along the curve of the water lines a rough rule can be used which will give a very close approximation to the actual distance, thus:

No. of Beams to Length.	Addition per Foot of Length.
3	·027ft.
4	·019ft.
5	·014ft.
6	·010ft.
7	·007ft.
8	·004ft.

The interval between Kriemhilda's frames is 5ft., and she is approximately five beams to length, then the addition per feet of length will be ·014ft., which multiplied by the interval (5ft. × ·014 = ·07ft.) will show the addition to be made to the interval to be ·07ft., making it 5·07ft.

Calculation of the Centre of Lateral Resistance.

This is a very simple calculation to make so far as it goes, and is done wholly from the sheer plan; but, as already explained on page 70, the calculations from the sheer plan do not give the actual position of the centre of lateral resistance; the point can, however, be ascertained from a model made to scale from the drawing. Fix a small nail at some point in the side of the vessel below the L.W.L.; to this nail attach a piece of string, and by it tow the model broadside on. If the nail is at the centre of the model, the latter will tow without showing a disposition to turn; in other words, the keel will keep at right angles to the direction of the towing strain. This experiment, if properly conducted, would give results sufficiently accurate to indicate how far the actual centre of gravity would be from the calculated centre in certain models; but, of course, it would not afford any clue as to the point the centre would shift to when the vessel became much inclined, and moved ahead (as well as broadside on) at great velocities.

The ordinates are measured on the sheer plan from the load water-line to the line representing the underside of the keel at the stations numbered 1½, 2, 2½, 3, 4, 5, 6, &c., and entered in their proper column as shown overleaf. They are then operated upon by Simpson's multipliers, according to the method already described.

Note.—In the case of a vessel with a fore foot like Minerva, or with a much less pronounced curve than shown by Kriemhilda, the ordinates at half intervals need not be used.

The whole of the rudder, indeed every portion of the immersed longitudinal section, must enter into this calculation (see Plate X.).

It will be noted that extra ordinates have been put in between No. 1, No. 2, and No. 3 ordinates, and between No. 15, No. 16, and No. 17 ordinates, numbered respectively 1½, 2½, and 15½ and 16½. The half ordinates have been put in more with a view to familiarising the beginner with their use than for any necessity of their being present to insure accuracy in the calculations. For ordinary purposes ordinates 5ft. apart in a figure of such form and dimensions as that of Kriemhilda's longitudinal section will yield sufficiently accurate results, and the ordinates at half intervals can be safely dispensed with. The Trapezoidal Rule should not be taken for this calculation, but either Simpson's first or second rule.

The centre of lateral resistance may be found by experiment, thus: Take a clean piece of thin deal and shape it like the sheer plan of the vessel below the L.W.L., including the rudder; suspend this figure from two different points by means of a piece of string, and the direction of the

string, formed into lines across the figure will give the centre of gravity of their intersection (*see* page 332).

Nos. of Ordinates.	Lengths of Ordinates.	Simpson's Multipliers.	Products for Summation.	Multipliers.	Products or Moments.
1	0·00		0·00	0	0·00
$1\frac{1}{2}$	6·60	2	13·20	$\frac{1}{2}$	6·60
2	8·10	1	8·10	1	8·10
$2\frac{1}{2}$	8·8	2	17·60	$1\frac{1}{2}$	26·40
3	9·25	$1\frac{1}{2}$	13·87	2	27·74
4	9·85	4	39·40	3	118·20
5	10·22	2	20·44	4	81·76
6	10·60	4	42·40	5	212·00
7	11·00	2	22·00	6	132·00
8	11·30	4	45·20	7	316·40
9	11·55	2	23·10	8	184·80
10	11·80	4	47·20	9	424·80
11	12·00	2	24·00	10	240·00
12	12·05	4	48·20	11	530·20
13	12·10	2	24·20	12	290·40
14	12·05	4	48·20	13	626·60
15	11·80	$1\frac{1}{2}$	17·70	14	257·80
$15\frac{1}{2}$	11·20	2	22·40	$14\frac{1}{2}$	324·80
16	10·30	1	10·30	15	154·50
$16\frac{1}{2}$	8·80	2	17·60	$15\frac{1}{2}$	272·80
17	6·60	$\frac{1}{2}$	3·30	16	52·80

```
508·41                      4278·70 × ½ 5ft.
     5                            5
3)2542·05                3)21393·50
   847·35         847·35)7131·166)8·4158*
                         6778·80
                          352·366
                          338·940
                           13·4260
                            8·4735
                            4·95250
                            4·23675
                             ·715750
                             ·677880
                             ·037870
```

*8·4158 × 5ft. interval.
5

42·0790ft. That is, the centre of gravity of the section, or centre of lateral resistance, as it is termed, is 42·0790ft. abaft the fore side of the stem.

If it is considered necessary to take the measurements in the manner described on page 71, &c., they will be operated upon in a precisely similar manner, as shown in the example just given.

The mean draught or depth of the Kriemhilda will be the area of the immersed vertical longitudinal section (shown by the sheer plan) divided by the length of the figure—that is, its length from No. 1 section at the stem to No. 17 section, which passes through the rudder. There are 16 intervals of 5ft., therefore $16 \times 5 = 80 =$ the length of the figure.

Then area = 848 sq. ft. ÷ length = 80ft = 10·6ft. = mean depth or draught of water.

$$\text{Mean draught} = \frac{848}{80} = 10{\cdot}6$$

The position of the centre of lateral resistance below the load water-line should be found to render the calculations complete, although in practice this position need seldom be determined.

To carry the calculation out the length of each water-line must be measured on the sheer plan from the fore side of the stem to its termination in the rudder. These lengths will be operated upon by Simpson's First Rule, and their moments found as shown in the example to follow. The sum of these moments will then be divided by the area of the section (already found); the quotient multiplied by the distance the water-lines are apart will give the distance the centre of lateral resistance is below the load water-line. An extra line below No. 7 will have to be put in the sheer plan at its proper distance, viz., 1·5ft. below No. 7. No. 9 will be formed by the underside of the keel measured 1·5ft. below No. 8. It will be seen that the numerical multipliers 1, 2, 3, 4, &c., are the numbers of the respective water-lines from the water-line downwards.

Nos. of Water-lines.	Lengths of the Lines.	Simpson's Multipliers.	Products to be Summed.	Multipliers.	Moments to be Summed.
L.W.L. No. 1	80·0	1	80·0	0	0·0
2	79·9	4	319·6	1	319·6
3	79·7	2	159·4	2	318·8
4	79·2	4	316·8	3	950·4
5	78·4	2	156·8	4	627·2
6	75·6	4	302·4	5	1512·0
7	69·0	2	138·0	6	828·0
8	50·0	4	200·0	7	1400·0
9	10·0	1	10·0	8	80·0

6036·0
× ½ interval 1·5ft. 1·5

3018·00
6036·0

3)9054·00

÷ area = 848)3018·00(3·55*
2544

474·0
424·0

50·00
42·40

7·60

* 3·54 × 1·5ft. interval.
1·5

1·775
3·55

5·325

That is, the centre of lateral resistance is 5·325ft. below the load water-line. As a rule, in practice, it will be unnecessary to make this

calculation, as it can be assumed that *the centre is half the mean draught below the load water-line* (see on the previous page).

THE META-CENTRE.

The calculation of the position of the meta-centre above the centre of buoyancy is usually made for all ships in conjunction with that for determining the centre of buoyancy, &c., but in reality it conveys but little meaning, and has but small value unless the position of the centre of gravity of the ship be known. However, as the metra-centric height generally figures in all calculations, the manner of its calculation is introduced at this point. The object of knowing its position will be more fully explained hereafter in the article on stability. Upon reference to pages 25, 26, &c., the diagrams show the value of knowing the position of the meta-centre. The mode of calculating its position for an indefinitely small angle of heel is a very simple one.

The mathematical expression for determining its height is :

$$\text{Height of meta-centre} = \frac{\frac{2}{3}\int y^3\, d\,x.}{\text{Displacement.}}$$

Where y represents any ordinate of the curve of the load water-line, and $d\ x$ any interval of length between the ordinates; then the ordinates (or, in other words, the various half breadths of the load water-line) having been cubed, are next summed by Simpson's First Rule (page 323). Two-thirds of the area thus found divided by the whole displacement (expressed in cubic feet) will give the height of the meta-centre above the centre of buoyancy.

The half breadths of the Kriemhilda's load water-line are taken from the table on page 339; each half breadth is cubed, and, in order to save time and labour, the cubes of the various half breadths will be taken from a table of cubes. Thus, the second ordinate of the Kriemhilda's load water-line is set down as 1·35ft.; upon reference to a table of cubes we find that the cube of 135 is 2460375, and thus the cube of 1·35 will be 2·460375; in selecting the cubes only one place for decimals will be retained in this calculation, so all the figures to the right of the first decimal will be struck off. The various half breadths having been treated in this way they will be summed as before explained by Simpson's First Rule, but from this point the calculation here given will best speak for itself. In general practice it will be quite safe to neglect the multipliers, and sum the cubes of the ordinates by the rule given on page 324.

The displacement of the Kriemhilda is 115·06 tons, which reduced to cubic feet = 115·06 × 35 = 4027.

Distinguishing Nos. of Ordinates.	Lengths of Ordinates in linear feet.	Cubes of Lengths of Ordinates.	Simpson's Multipliers.	Products of Cubes.
1	0·20	0·0	1	0·0
2	1·35	2·4	4	2·4
3	2·75	20·7	2	41·4
4	4·20	74·0	4	296·0
5	5·60	175·6	2	351·2
6	6·75	307·5	4	1230·3
7	7·65	447·6	2	895·2
8	8·30	571·7	4	2286·8
9	8·60	636·0	2	1272·0
10	8·70	658·5	4	2634·0
11	8·60	636·0	2	1272·0
12	8·28	567·6	4	2270·4
13	7·90	493·0	2	986·0
14	7·05	350·4	4	1401·6
15	5·35	153·1	2	306·2
16	2·70	19·6	4	78·4
17	0·30	0·0	1	0·0

15330·8
× $\frac{1}{3}$ 5ft. the distance the ordinates are apart. 5

3)76654·0

Divided by displacement...... 4027)25551·3(6·344*
24162

1389·3
1208·1

181·20
161·08

20·120
16·108

4·012

* $\frac{2}{3}$ of this quotient, 6·344, will be found thus 6·344
2

3)12·688

4·229ft. = height the meta-centre is above the centre of buoyancey.

The Wedges of Immersion and Emersion.

The calculations for determining the volumes of the wedges of immersion and emersion—or the "in" and the "out" wedges, as naval architects term them, will next be given. The necessity of making these calculations need not be dwelt upon here, as the subject will be considered in another portion of the book.

To the inexperienced the calculations will no doubt look difficult

and complex, but in reality they are no more difficult than the calculation of displacement, although, perhaps, they are a little more tedious. Upon reference to the body plan of the Kriemhilda (Fig. 159) it will be found that the vessel is supposed to be heeled to an angle of 20°. The area of that portion of each vertical section which has been immersed or emersed by inclination of the vessel has to be found by the rule explained on pages 326 and 327.

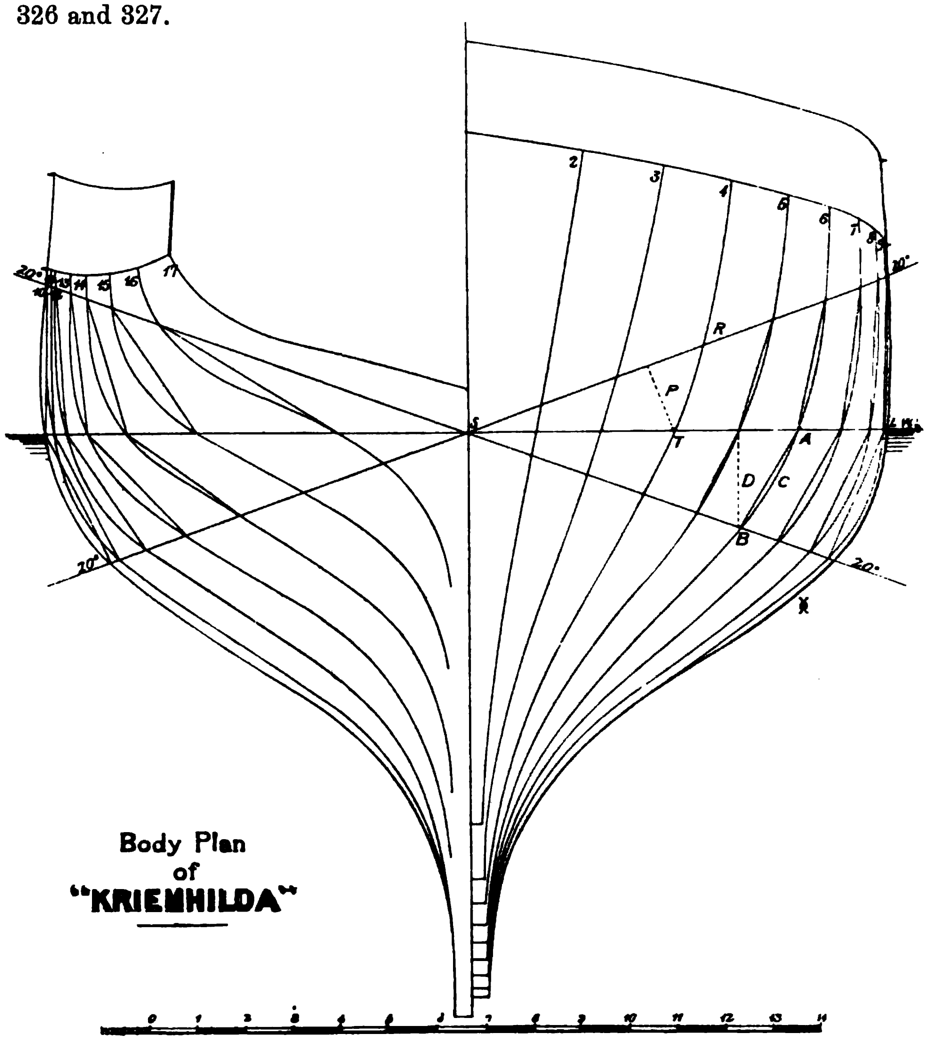


FIG. 159.

That is, the parabolic segment must be cut off each section, and its area and the area of the triangle computed separately, according to the rules referred to.

Fig. 159 shows the body plan of Kriemhilda prepared for this operation. Taking the immersed wedge first (which will be the portion above

the L.W.L.) we find that the frames up to No. 5 section are sufficiently straight to admit of the figure being treated as a triangle. Take No. 4 section as an example; R S T are the three sides of the triangle; P is set off at right angles to R S, and R S will be called the base and P the perpendicular; then (*see* pages 325-327) the distance multiplied by half the distance P will give the area of the triangle R S T.

The triangle for the emersed wedge, No. 6 section, will illustrate the calculation, which includes the parabolic segment. The segment A B C has been cut off, leaving the triangle A B S to be calculated as just explained, D being the perpendicular of the triangle. The area of the segment A B C is calculated by the rule given on page 327.

Upon reference to the body plan (Fig. 159) it will be found that there is an outside section next to No. 9 marked ⊗, this is the midship section, and is not considered in the calculations about to be made.

Immersed Wedge.

Section 1.

There is no area for No. 1 section, it being formed by the half-siding of the stem.

Section 2.

Triangle.
1·55ft. = base of triangle.
·45ft. = perpendicular.

775
·620

2) ·6975

·3487 = area.

There was no parabolic segment to this area nor to any of the other until No. 6 is reached.

Section 6.

Triangle.
7·80 = base.
2·30 = perpendicular.

2·3400
15·60

2)17·94

8·97
·18

9·15 = area.

Parabolic Segment.
2·70 = base.
·10 = height.

·2700
2

3) ·5400

·18 = area.

The other sections are similarly calculated.

In order to arrive at the volume of the wedge the respective areas must be tabulated and summed by Simpson's First Rule, and the centre of gravity will be found by the same process.

The method has been varied to illustrate another plan of tabulating the calculations.

Volume of the Immersed Wedge.

Nos. of Sectional Areas.	Areas of Sections.	Multipliers.	Moments.
1	0·00	0	0·00
2	0·35	1	0·35
3	1·47	2	2·94
4	3·78	3	11·34
5	6·34	4	25·36
6	9·15	5	45·75
7	11·42	6	68·52
8	12·73	7	89·11
9	13·48	8	107·84
10	13·78	9	124·02
11	13·51	10	135·10
12	12·80	11	140·80
13	11·66	12	139·92
14	10·51	13	136·63
15	8·32	14	116·48
16	3·82	15	57·30
17	0·00	16	0·00

Areas of Even Nos. of Sections.

2 ...... 0·35
4 ...... 3·78
6 ...... 9·15
8 ...... 12·73
10 ...... 13·78
12 ...... 12·80
14 ...... 10·51
16 ...... 3·82

66·92
4 Simpson's multiplier.

267·68
132·40

400·08 × ⅓ interval of 5ft.
5

3)2000·40

666·80 = volume of the wedge of immersion.

Areas of Odd Nos. of Sections.

3 ...... 1·47
5 ...... 6·34
7 ...... 11·42
9 ...... 13·48
11 ...... 13·51
13 ...... 11·66
15 ...... 8·32

66·20
2 Simpson's multiplier.

132·40

Centre of Gravity of the Volume of Immersion.

Even Nos. of Moments.

2 ...... 0·35
4 ...... 11·34
6 ...... 45·75
8 ...... 89·11
10 ...... 120·02
12 ...... 140·80
14 ...... 136·63
16 ...... 57·30

605·30
4

2421·20
1192·32

3613·52 × ⅓ interval of 5ft.
5

3)18067·60

÷ by volume 666·80)6022·53(9 × 5ft.
6001·20 5

21·33 45 = the distance the centre of gravity of the immersions is abaft the fore side of the stern.

Odd Nos. of Moments.

3 ...... 2·94
5 ...... 25·36
7 ...... 68·52
9 ...... 107·84
11 ...... 135·10
13 ...... 139·92
15 ...... 116·48

596·16
2

1192·32

Volume of the Emersed Wedge.

The emersed wedge is that portion of the vessel which is taken out of the water upon inclination of the vessel, and the calculation to determine its volume and its centre of gravity does not differ from that of the volume of immersion.

Volume of Emersed Wedge.

Nos. of Sectional Areas.	Areas of Sections.	Multipliers.	Moments.
1	0·00	0	0·00
2	0·30	1	0·30
3	1·17	2	2·34
4	2·73	3	8·19
5	4·78	4	19·12
6	6·90	5	34·50
7	8·98	6	53·88
8	10·76	7	75·32
9	11·70	8	93·60
10	12·01	9	108·09
11	11·62	10	116·20
12	10·49	11	115·39
13	8·89	12	106·68
14	6·38	13	82·94
15	3·48	14	48·72
16	0·84	15	12·60
17	0·00	16	0·00

Areas of Even Nos. of Sections.

2 ...... 0·30
4 ...... 2·73
6 ...... 6·90
8 ...... 10·76
10 ...... 12·01
12 ...... 10·49
14 ...... 6·38
16 ...... 0·84

50·41
4

201·64
101·24

302·88 × ⅓ interval of 5ft.
5

3)1514·40

504·80 = volume of emersion at 20° inclination.

Areas of Odd Nos. of Sections.

3 ...... 1·17
5 ...... 4·78
7 ...... 8·98
9 ...... 11·70
11 ...... 11·12
13 ...... 8·89
15 ...... 3·48

50·92
2

101·24

Centre of Gravity of the Volume.

Even Nos. of Moments.

2 ...... 0·30
4 ...... 8·19
6 ...... 34·50
8 ...... 75·32
10 ...... 108·09
12 ...... 115·39
14 ...... 82·94
16 ...... 12·60

437·33
4

1749·32
881·08

2630·40 × ⅓ interval 5ft.
5

3)13152·00

÷ by volume 504·80)4384·00(8·68 × 5ft. interval.
4038·40 5

345·600 43·40ft. = the distance the centre of gravity of the volume of emersion is abaft the fore side of the stem.
302·880

42·7200
40·3840

2·3360

Odd Nos. of Moments.

3 ...... 1·34
5 ...... 19·12
7 ...... 53·88
9 ...... 93·60
11 ...... 116·20
13 ...... 106·68
15 ...... 48·72

440·54
2

881·08

The method of calculating the volume of the wedges and their centre of gravity by polar co-ordinates will be given further on in the calculations for stability.

CALCULATION OF STABILITY.

Centre of Gravity of the whole Mass.

The centre of gravity of a ship can only be approximated to by calculation before she is built; but that point can be exactly determined by experiment after she is in the water.

It is already known that in the upright position the centre of gravity lies somewhere in the line V F M (Fig. 160), and its exact distance below M can be found by experiment.

Let F (Fig. 160) be the centre of gravity (not to be confused with centre of buoyancy) of the vessel, M the meta-centre, and O any heavy weight forming a part of the vessel's equipment; by moving O to a

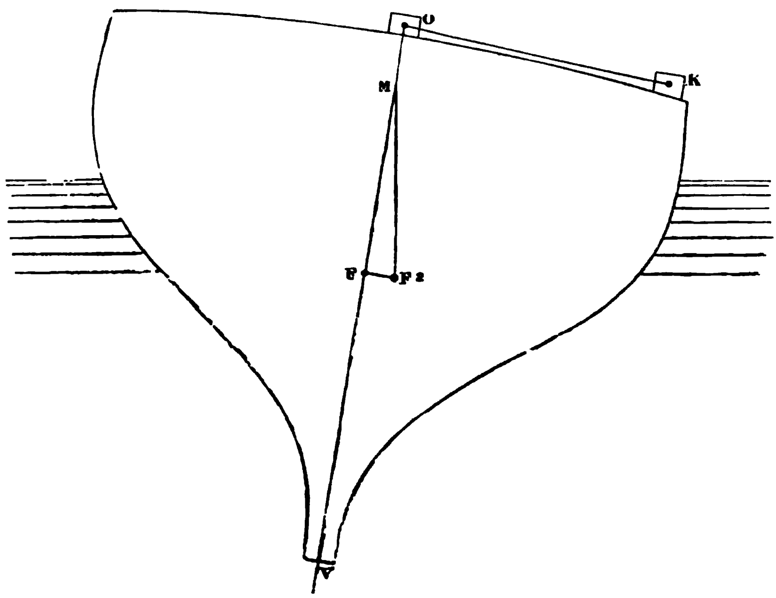


Fig. 160.

different position on one side to K, the vessel will be inclined, in consequence of the centre of gravity being shifted transversely with the shifting of the weight, to some point, as F_2; the centre of gravity will thus now be in the line M, F_2 having shifted in a direction parallel to O K and less than O K, as the weight shifted is less than the whole weight of the vessel.

In the example now given the Sea Belle schooner was the vessel experimented upon. A long plank of deal was placed across the vessel upon the bulwarks, shown by A A (Fig. 161). The plank was properly

secured by lashings, and was placed as nearly as possible in the midship part of the vessel.

A weight of ballast, consisting of oblong bricks of lead, equal to half a ton in the aggregate, was then put on the middle of the plank, W, having been taken from below the platform or lower deck; the height of the plank above the water was then measured on either side, and the weight adjusted until each end of the plank was equidistant from the water, and the vessel proved to be nearly in an upright position.

Of course the plank need not always be used, as the weight might be heavy enough to give sufficient heel by moving it across the deck only from the centre to near the bulwarks.

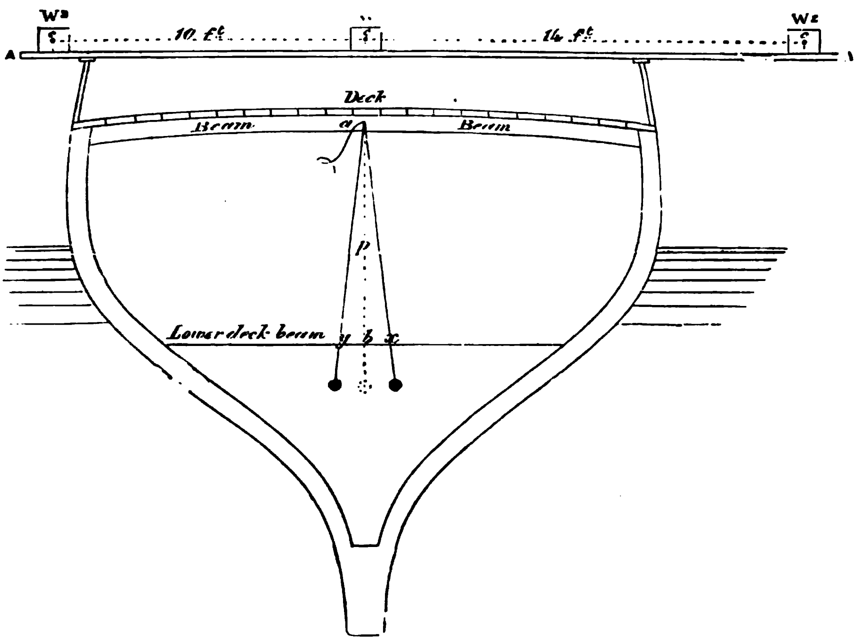


FIG. 161.

If a pile of pig ballast be used, or any other weight, it must be re-stowed when it has been removed in exactly the same way it was when in the centre of the deck, and the distance it has been moved will be measured from the centre of the weight before removal to the centre after removal.

In the main cabin a line with heavy plumb attached, represented by the ticked line P, was let fall from a deck beam at the point *a*, and whilst the vessel was quite still the point where the line cut the lower deck beam was accurately marked at *b*. A lower deck beam is not always available, and a good plan is to have a bucket of water with a bar of wood lashed across it to the handle. This bar would be marked in the same manner as the beam, and the water would keep the plumb steady.

The weight, W, was then shifted to W_2, 14ft.; the distance 14ft. being arrived at by measuring from the centre of the weight c to the centre of the weight c when shifted. This distance will be called k. The distance (x) the plumb line was deflected along the lower deck beam from b was then carefully measured and found to be $1\frac{1}{8}$in., and the length of the plumb line, $a\ x$, 6ft. $10\frac{1}{4}$in. The weight W was then shifted 10ft. in the opposite direction, to W_3; the result was that the plumb line was deflected $\frac{13}{16}$in. $= b\ y$.

The centre of gravity of the vessel below M will now be determined (see Fig. 161) thus:

$$\text{M F} = \left(\frac{\text{W} \times \text{distance } k}{\text{Displacement.}}\right) \text{cotangent } \theta$$

That is, 14ft., the distance the weight (W) was shifted, has to be multiplied by that weight (= half a ton); the product will be divided by the displacement of the vessel expressed in tons; the quotient multiplied by the cotangent of the angle of heel will give the distance the centre of gravity is below the meta-centre. The distance equals 14ft., the weight shifted was 0·5, or half a ton, the displacement or weight of the vessel was 155 tons, then

$$\begin{array}{l}
\qquad\qquad\qquad\qquad\qquad\quad 14 \quad \text{distance} \times 0\cdot5 \text{ weight.} \\
\qquad\qquad\qquad\qquad\qquad\quad \underline{0\cdot5} \\
\text{Displacement tons } 155)7\cdot00(00\cdot4516 \\
\qquad\qquad\qquad\qquad\qquad\quad \underline{6\cdot20} \\
\qquad\qquad\qquad\qquad\qquad\quad\ \ \cdot800 \\
\qquad\qquad\qquad\qquad\qquad\quad\ \ \underline{775} \\
\qquad\qquad\qquad\qquad\qquad\qquad 250 \\
\qquad\qquad\qquad\qquad\qquad\qquad \underline{155} \\
\qquad\qquad\qquad\qquad\qquad\qquad\ \ 950 \\
\qquad\qquad\qquad\qquad\qquad\qquad\ \ \underline{930} \\
\qquad\qquad\qquad\qquad\qquad\qquad\quad \underline{20}
\end{array}$$

This quotient, 0·04516 (representing the distance the centre of gravity has been shifted), has to be multiplied by the cotangent of the angle of heel.

To find the cotangent of the angle of inclination, divide the length of the plumb line $a\ x$ (Fig. 161) by the distance it was deflected: $\frac{6\cdot854}{\cdot09374}$

$$\begin{array}{l}
\cdot09374)6\cdot8540(73\cdot117 = \text{cotangent of } 0^\circ\ 47' \\
\qquad\quad\ \underline{6\cdot5618} \\
\qquad\qquad 29220 \\
\qquad\qquad \underline{28122} \\
\qquad\qquad\ \ 10980 \\
\qquad\qquad\quad \underline{9374} \\
\qquad\qquad\quad 16060 \\
\qquad\qquad\quad\ \ \underline{9374} \\
\qquad\qquad\qquad 66360 \\
\qquad\qquad\qquad \underline{65618} \\
\qquad\qquad\qquad\ \ \underline{1242}
\end{array}$$

The distance the centre of gravity was shifted = ·04516 will be multiplied by 73·117 = cotangent of 0° 47′.

$$\begin{array}{r} 73{\cdot}117 \\ \underline{{\cdot}04516} \\ 038702 \\ 73117 \\ 365585 \\ \underline{2{\cdot}92468} \\ \underline{3{\cdot}30196372} \end{array}$$

That is, the centre of gravity of Sea Belle is 3·3ft. below the meta-centre; a correction, however, due to lifting the weight from the bottom of the vessel has yet to be made. This correction will be explained below.

A second experiment should be made to verify the first. In this case W was shifted 10ft. to W 3 and the plumb was deflected $\frac{13}{16}$in. = ·06771ft., and the length of the plumb line was 6·85ft.; then cotangent $\theta = \frac{6{\cdot}85}{{\cdot}06771}$ = 101· = cotangent θ 0° 34′; or Sea Belle was inclined a little over half a degree by shifting the weight 10ft.

Proceeding as before $\frac{10\text{ft.} \times 0{\cdot}5}{155}$ = ·03226. The cotangent of 0° 34′ is 101·, then ·03236 × 101· = 3·26 = the distance the centre of gravity is below the meta-centre.

Taking the mean of the two results:

3·30 ...... by shifting the weight 14ft.
3·26 ...... by shifting the weight 10ft.

2)6·56

3·28 = the distance the centre of gravity is below the meta-centre.

As before stated a correction has to be made for lifting the half ton of lead ballast from the other ballast. From the top side of the plank to the top of the ballast from whence the half ton was taken the distance was 13ft. 5in.; then (see page 331) the weight of half a ton must be multiplied by the distance that weight has been lifted, and divided by the whole weight (or displacement) of the vessel, thus:

13·5ft. distance the weight was lifted.
·5 ton, the weight.

Displacement 155)6·75(0·0435ft.

$$\begin{array}{r} \underline{6{\cdot}20} \\ 550 \\ \underline{465} \\ 850 \\ \underline{775} \\ 75 \end{array}$$

This quotient shows that the centre of gravity of the vessel would have been ·0435ft. lower if the half ton of ballast had been in its proper place; herefore, ·0435ft. must be added to the distance the centre of gravity is below the meta-centre as previously determined, thus: 3·28 + ·0435 =

3·3235ft. = the exact distance the centre of gravity of the Sea Belle is below her meta-centre.

In making the experiment to determine the position of the centre of gravity the vessel must be in perfectly still water, and there should be no wind to cause her to heel, or much tide to cause her to sheer and strain on her cable. After the plumb line has been fixed, and after the weight has been shifted, everyone on board should remain quite still whilst the person who is making the experiment watches the deflection of the plumb line. The vessel should not be made fast by warps, but be free to oscillate. The sails must of necessity be stowed, but the alteration in the position of the centre of gravity by their being hoisted can be readily calculated if their weight is known. As a set-off against the sails, gaff, &c., when hoisted, the weight of hemp and chain halyards coiled on deck will have to be considered, and in the end the calculation would give some trouble and be practically of incommensurate value. Unless the person who conducts the experiment has had some experience, it would, in order to avoid inaccuracy, be better to employ a weight that would heel the vessel to three or four degrees.

Calculating the Centre of Gravity by the Weights of Material.

The centre of gravity of Sea Belle was calculated by dealing with all her weights separately. With experience and care the centre of gravity of a vessel can very closely be approximated to in this way, but the calculations would be worse than useless if carelessly made. The sizes of the frames and all the scantling must be known. Then the weight of every portion of the vessel's hull and equipment must be calculated separately, and the distance the centre of gravity of each portion of the hull, &c., must be found relative to some fixed point. This point, in the example about to be given for the Sea Belle, was placed at the load water-line, so the weights are divided into those "below" and above the load water-line; but it is more usual to measure the distance the centre of gravity of each part of the vessel is from a base line drawn, say, on the under side of the keel, and parallel to the load water-line.

In the case of sea Belle, on measuring the frames from the load water-line to the dead wood, the mean length was found to be 11ft. The mean siding of the frames is 0·46ft., and the mean moulding 0·35ft. Then $11 \times \cdot 46 \times \cdot 35 = 1\cdot 77$ = the cubical contents of each frame. There are eighteen such frames on each side of the vessel, or thirty-six in all, then $1\cdot 77 \times 36 = 64$ = cubical contents of the frames. One cubic foot of English oak weighs 48lb. and 2240lb. go to the ton; then $64 \times 48 =$ 3072lb. = 1·37 ton.

The centre of gravity of the frames was calculated to be half the mean depth measured perpendicularly from the load line to the dead wood.

The intermediate frames are 11 × ·33 × ·25 = 0·91 cubic feet. There are seventy-two of these frames, then 72 × 91 = 65·5 cubic feet = 1·4 ton. The centre of gravity of these will lie at the same point as the centre of gravity of the other frames.

The keel is 0·8ft. sided, 1ft. moulded, and from the load water-line round the fore foot to the stern post measures in length 90ft.; then 90 × 1 × ·8 = 72 cubic feet = 1·6 ton.

To arrive at the cubical contents of the plank, the following plan will be adopted: mean length of the water-lines (measured on the half breadth plan) × mean girth of frames. The mean length of the water-lines is 88ft., and mean girth of frames 11ft.; then 88 × 11 = 968. The thickness of the planking is 3½in.= ·3ft.; then 968 × ·3 = 290 cubic feet, or for both sides of the vessel 580 cubic feet = 12·4 tons.

The bulk of the inside keel and keelson of wood, the dead wood stern post (measured from the *deck*), rudder, platform, beams, &c., and inside fittings, are all calculated and entered in the tables.

The lead keelson must be a separate calculation: 20·5ft. of its length is ·75 × ·75 = 11·5 cubic feet; 5·4ft. of its length at the fore end, ·75ft. × ·4ft. = 1·6 cubic feet, making 13 cubic feet in all. There are 3·16 cubic feet to one ton of lead; then $\frac{13}{3 \cdot 16}$ = 4·1 tons = weight of lead keelson.

The lead lumps between the floors are all ·75 × 1·16ft., but vary in depth. Their bulk must be calculated in detail, but it is unnecessary to give the figures here. In the aggregate the lumps contain 44·24 cubic feet, exactly equal to 14 tons.

The weight of each lead floor was also calculated separately, and found in the aggregate to equal 19·7 tons; whilst the centre of gravity of the whole of the same lay 6·8ft. below the load water-line.

The portions of the hull of the vessel above water were calculated in a similar way, and set out in the first column of the table.

In the second column the centre of gravity of each item ($\frac{\text{above}}{\text{below}}$) the load water-line is entered. The weight of each item is then multiplied by vertical distance its centre of gravity is from the load water-line; the products given are the moments of the weights. The moments are then summed, and the sum of those above the load water-line are subtracted from those below. The remainder is then divided by the weight in tons of the vessel, and the quotient is the distance the centre of gravity is below the load water-line. In the example given, the calculated centre of gravity was 2·6ft. below the load-water line; by experiment, the centre of gravity was proved to be 2·8ft. below, so that the error was on the safe side.

Weights above the Load Water-line.

	Tons.	Centre of Gravity above L.W.L.	Moments.
Weight of frames	0·6	2·1	1·5
" plank	5·0	2·1	10·5
" stem knightheads, &c.	0·5	5·0	2·5
" deck beams	1·4	3·8	5·3
" deck plank	4·0	4·1	16·4
" stern frames	0·8	4·0	3·2
" stanchions	0·4	5·0	2·0
" bulwarks	0·5	5·0	2·5
" rail	0·4	6·0	2·4
" shelf	0·3	3·8	1·1
" deck fittings	2·5	5·0	12·5
" capstan	0·4	7·0	2·8
" masts, including topmasts and ironwork crosstrees	5·3	30·0	159·0
" bowsprit, including ironwork	1·0	8·0	8·0
" main boom	1·7	8·0	13·6
" gear, spare sails, &c.	4·0	0·0	0·0
" gaffs (stowed)	1·0	11·0	11·0
" rigging and blocks	3·0	20·0	60·0
" sails (stowed)	2·3	10·5	24·1
" hammocks, &c.	0·5	2·0	1·0
" crews and other persons on deck	1·5	6·0	9·0
" davits	0·4	4·0	1·6
" two boats on deck	0·8	5·5	4·4
	38·3		354·1*

Weights below the Load Water-line.

	Tons.	Centre of Gravity below L.W.L.	Moments.
Weight of frames	1·7	4·5	7·6
" intermediate bent timbers	1·3	4·5	5·8
" keel	1·6	11·0	17·6
" plank	12·4	4·5	55·8
" dead wood at bow	0·5	4·5	2·2
" " at stern	0·6	7·0	4·2
" inside keel of wood	1·0	10·5	10·5
" keelson of wood	0·5	7·5	3·7
" stern post	0·3	7·3	2·2
" rudder	0·4	5·2	2·1
" beams for platform, and platform	1·3	4·5	5·8
" inside fittings	4·5	0·5	2·2
" lead on keel	6·4	11·7	74·9
" lead keelson	4·1	7·4	30·3
" lead lumps between floors	14·0	8·7	122·0
" lead floors	19·7	7·5	147·7
" limber ballast	21·6	7·0	151·2
" iron mast step	0·9	6·5	5·8
" other ballast, loose	6·3	6·0	37·8
" anchors and chains (stowed)	3·7	4·5	16·6
" galley and cabin stove	0·2	0·6	0·1
" two tanks, two tons of water	2·7	4·2	11·3
" coal and coke	1·0	5·0	5·0
" stores, furniture, &c.	2·6	0·0	0·0
" warps	1·0	0·0	0·0
" iron, metal bolts, fastenings, &c., in hull	3·8	6·0	22·8
" copper	2·4	4·6	11·0
	116·5		756·2†

† 756·2
* 354·1

155)402·1(2·59 = the distance the centre of gravity is below the load water-line.

[The following is a specimen of the method for calculating the centre of gravity of an 80-tons cutter by Mr. Ardagh E. Long.

80-Tons Racing Cutter—Weight and Centre of Gravity.

	Weight Tons.	Lever, Ft.	Moments.
Stem bar	0·92	8·00	7·36
Sternpost	0·36	7·93	2·85
Rudder complete	0·73	8·42	6·14
Shell plating, including keel plate	16·14	10·76	173·67
Frames	3·68	10·90	40·12
Reverse frames	1·09	6·54	7·13
Doubling on frames in way of ballast	0·40	1·94	0·77
Upper deck beams	0·75	18·80	14·10
Lower deck beams and knees	0·52	10·36	5·43
Diagonal struts	0·18	8·67	1·56
Floor plates	1·01	3·21	3·24
Upper deck and fastenings	3·73	18·98	70·79
Lower deck wood and fastenings	1·34	10·66	14·28
Upper deck stringer and angle	2·10	19·05	40·00
Tie plates, longitudinal and diagonal, mast plate, &c.	0·59	19·06	11·25
Gusset plates on struts amidships	0·09	10·02	0·90
" " " at sides	0·16	6·64	1·06
Bulwark plating and stanchions	0·84	19·95	16·75
Counter ridge	0·06	17·72	1·06
Ceiling in 'tween decks	1·75	14·91	26·09
" hold	1·80	6·66	11·99
Cabin fittings and forecastle fittings	4·00	14·02	56·08
Deck fittings	2·20	20·18	44·39
Rail, including angle	0·63	20·55	12·95
Capstan	0·40	22·00	8·80
Freshwater tank, 500 gallons	2·75	4·30	11·82
Coal	1·00	4·65	4·65
Warps	0·50	8·50	4·25
Anchors and chains stowed	1·76	8·00	14·08
Stores	1·50	6·66	9·99
Spare gear and sails	1·70	6·66	11·32
Crew on deck	1·20	21·80	26·16
Boat on deck	0·20	20·48	4·09
Mast (lower) and fittings	1·95	40·32	78·62
Topmast housed	0·32	53·71	17·19
Boom and fittings	1·55	23·85	36·97
Gaff (aloft)	0·48	78·70	37·77
Bowsprit and fittings	0·80	22·80	18·24
Spinnaker boom on deck	0·51	18·50	9·44
Sails—mainsail, foresail, and jib set	1·20	46·71	56·05
Standing rigging and dead eyes	1·01	38·19	38·57
Running rigging	1·14	33·69	38·40
Blocks	0·50	42·00	21·00
Ballast fixed (including cement, 1·21 tons)	82·00	1·93	158·26
" loose in blocks	4·00	3·20	12·80
Pillars	0·25	14·91	3·73
Channels (all steel)	0·35	17·80	6·23
Paint, &c.	0·99	11·15	1·00
Topsail yards on deck	0·55	18·60	10·23
Reverse frames to gunwale in way of rigging	0·11	14·20	1·56
Doubling of keel plate	0·29	0·72	0·21
Woodwork in hold (fitting for stores, &c.)	0·50	7·20	3·60
	153·68	......	1216·99
			7·919ft.

Centre of gravity $\frac{1216 \cdot 91}{153 \cdot 68} = 7 \cdot 919$ft. above base line. Moments taken about a base line parallel to and 14·00ft. below L.W.L.].

Curve of Stability.

A vessel's stability can be progressively determined from her initial heel to any angle where her stability will vanish. These calculations were first introduced to the world by Mr. Barnes in a paper read at the Institution of Naval Architects in 1861; since that date the calculations have been more or less usefully employed, and attention was much directed to them by the loss of the Captain in 1870. In 1871 Sir W. H. White and the late Mr. W. John read a most elaborate paper at the Institution of Naval Architects on the calculation of curves of stability, and illustrated it with numbers of graphic diagrams; even more usefully than this, they gave the whole table of figures and sums in detail which had been employed in calculating the curve. These tables have proved most valuable, and have induced many who regarded the calculation of a curve of stability as a work only to be performed by a first-class mathematician to thoroughly master the problem of stability even so far as to determine its exact range. The compiler of this work is much indebted to these gentlemen for making the details of their labour known, as until the publication of the "Transactions of the Institution of Naval Architects" the formulæ current for determining stability were not readily adaptable for making curves of the same. By the system here introduced the exact stability to any angle of heel can be continuously calculated; and, although the mass of figures may at first appear overwhelming, the manner of using them will soon grow familiar.

It is necessary that the calculator should be provided with books of "mathematical tables," such as Barlow's and Law's; * without such tables the calculation of a "curve of stability" would prove an almost interminable labour.

The upright or vertical sections shown on the body plan (Fig. 162) are numbered 1, 2, 3, &c., No. 1 being the stem, and here represented by the middle vertical line. Each section terminates with the deck above and with the keel beneath the load water-line.

From the middle of the load water-line (L.W.L.) a number of lines radiate at equal angular intervals of 10°. The vessel is assumed to be heeled consecutively to all these angles up to 90°, when she is on her "beam ends." The exact stability at each angle of 10° interval is calculated.

Paper must be ruled to correspond with the tables for the "First Tables of Ordinates" of water sections.

The figures in the first column of the tables represent the numbers of

* Barlow's tables are published by Spon and Co., Charing Cross; Low's are in "Weale's Series," published by Crosby, Lockwood, and Co., Stationer's Hall-court.

the vertical sections, No. 1 being the stem. In the second column the lengths of the ordinates measured along the radiating lines to each vertical section are given.

It will be noted that at the next angle after 20° the deck is reached, and the radial lines will be measured to their intersection with the *horizontal* or deck lines corresponding with the vertical sections.

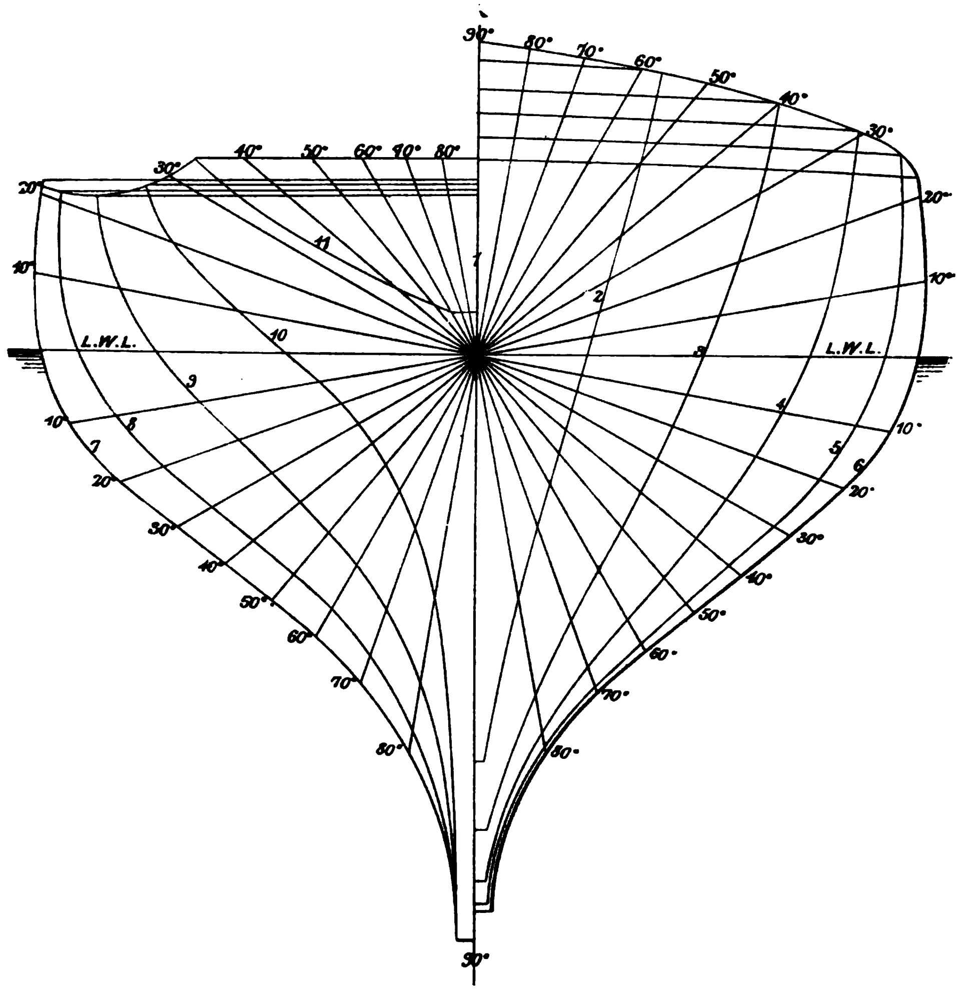


Fig. 162.

The ordinates, or radial lines, it will be seen, are multiplied by "Simpson's multipliers," and the products entered in the fourth column. These products are then added together, and divided by 3.

In the fifth column the *squares* of the ordinates, or radial lines, taken from a Table of Squares, are entered, and there multiplied by Simpson's multipliers. The products are added together and divided by 3.

In the eighth column the cubes of the ordinates, or radial lines, taken from a Table of Cubes, are entered, and, having been multiplied by Simpson's multipliers, their products are added together and divided by 3.

The first table represents the load water-line.

The second table relates to the water section formed by the vessel being heeled to 10°; the ordinates, or radial lines, are measured along the lines (on either side of the middle line of the body plan), making the angle of 10° with the water-line, and are then operated upon as just described. These ordinates form the upper side of the "immersed wedge" of 10° inclination, and the load water-line the lower or under side of the wedge.

The "emersed wedge" (or the wedge taken out of the water) has next to be operated upon. The lengths of the ordinates along the radial line 10° *below* the water-line are measured on either side of the middle line of the body plan, and being entered in the second column are operated upon exactly as were the ordinates of the immersed wedge.

The sum of the products of the cubes of ordinates of the *emersed* wedge are added to the sum of the products of *immersed* wedge.

The same process must be gone through for each angle of inclination given.

Tables of Stability.

For determining the stability fresh tables must be ruled in accordance with the examples given. As all these tables will be alike, that for 60° inclination will serve to illustrate the continuance of the work.

The first column is merely for entering the degrees of inclination. In the second column are entered the *products* of the simple lengths of the ordinates of the water section under treatment. In the third column the sum of the products of squares of ordinates for each inclination are put, and affected by multipliers.

The final products for each wedge thus found are then summed, and the greater sum deducted from the lesser; in this instance, and in fact in almost every case, the immersed wedge is in excess of the emersed, and consequently the products for the *emersed* wedge will be deducted from those for the *immersed* wedge. The remainder is then divided by 2, and multiplied by one-third the angular interval (in circular measure) of the radial planes (of course the quantity of the angular interval may vary according to the value of Simpson's or such other multipliers as are used). The product is then multiplied by the longitudinal interval between the vertical sections, and this product yields in cubic feet the excess in the volume of one wedge over another.

The reason of these operations will be better understood by referring to page 329. It should be noted that the divisor 2 is used because the

half squares only of the ordinates are required for calculating the volumes of the wedges and their differences.

We may here stop to again show how the volume of any wedge formed by the inclination of the vessel can be found by the use of the half squares of the ordinates. For example, take the stability table for 20° inclination: the sum of the products of the squares of the ordinates of the L.W.L.; of the water section at 10°; and the water section at 20° are operated upon according to the rule described, page 325: the sum of the products is 3186·4. This sum divided by 2 will give the sum of the half squares as required by page 329. The quotient will next be multiplied by one-third the angular interval in circular measure, and by the longitudinal interval. (It will be noted that the whole interval of 9ft., instead of one-third of that quantity, is used as the longitudinal multiplier; the reason is that the sums of the products of squares of ordinates had already been divided by 3 in the "first tables." But it is plain that time might have been saved, if, instead of dividing the sums of the products of the squares by 3, they had been multiplied by 3, that quantity being equal to the required one-third of the longitudinal interval of 9ft.) The volume of the wedge will then be determined as follows: $\frac{3186{\cdot}4 \times {\cdot}058 \times 9}{2} = 831{\cdot}6$ cubic feet = volume of the wedge.

In this calculation for stability it is not necessary to determine the quantity contained separately in the volumes of immersion and emersion; the excess in the one over the other is only required. It is necessary to know this in order to determine the volume and thickness of the layer *a a*, referred to in pages 25, &c., and a few pages farther on.

The volume of the layer for 60° inclination (see table of stability for 60° inclination) has been found to be 529·6 cubic feet (equal to the excess in the immersions): the thickness of that layer has now to be determined, and to determine this the area of the water section must be known. [The area of the water section here means the area of the plane of flotation at 60° inclination, just the same as the area of the load water-line is the area of the plane of flotation of the upright position.]

The area of the water section will be thus determined. In the "First Tables of Ordinates" for 60° inclination will be found (in the fourth column) the sum of the ordinates for the immersed section set down as 49·5, and for the emersed 51·0. These two sums are added together, and multiplied by 9ft., the longitudinal interval; the product is the area of the plane of flotation, or water section, at 60° inclination (*see* "Area of Inclined Water Section" in the "Tables of Stability" for 60°). The volume of the layer (*i.e.*, the excess of the immersions) is divided by the area of the water section, and the quotient is the thickness of the layer.

This thickness set off below at right angles to the given radial line will show where the line of the true water section, or plane of flotation, should be at the given angle of inclination (*see* "Layer" in the "Tables of Stability" for 60°). For this calculation of statical stability the thickness of the layer is not required. The object of knowing it will be shown when the dynamical stability of the vessel is considered.

The volume of the layer is used for statical stability to determine the extent of a correction that has to be made in the moments of stability, the use of which will hereafter be referred to. The transverse position of the centre of gravity of the inclined water section is thus found: The products of the squares of the ordinates are taken, and the lesser subtracted from the greater. The remainder shows the excess, and in this case for 60° the emersed section is in excess. The excess is divided by 2, and the quotient is multiplied by the longitudinal interval of 9ft. This last product is then divided by the area of the inclined water section, and the quotient shows how far the centre of gravity of the water section is on one side or the other of the middle line of the vessel, or the true point whence the ordinates shown in the body plan (Fig. 162) radiate. In the example for 60° it is proved that the centre of gravity of the inclined water section is on the emersed side, having changed from the immersed side between 50° and 60° inclination. Thus it is assumed, and practically it is correct enough, that the centre of gravity of the layer is in the vertical in which was found the centre of the water section. In very large vessels, or where the layer is very thick, the true centre would be calculated and used.

The volume of the layer multiplied by the distance the centre of gravity of the water section is from the middle line will give the correction that has to be made in the moments for statical stability. This correction is added or subtracted according to the following conditions: if the immersed *wedge* be in excess, and the centre of gravity of the *water section* lies towards the immersed side of the middle line, then the correction is subtracted; if the immersed *wedge* is still in excess (as in the case of 60°), and the centre of gravity of the water section lay on the emersed side of the middle line, then the correction is added. The object of making this correction is obviously on account of having introduced an excess in the moment, through not taking away the layer in the first instance, as explained, page 24.

To finally determine the statical stability the various "sums of the products of cubes of ordinates" of *both wedges* are taken from the "First Tables" and put in the sixth column of the "Tables of Stability;" they are there operated upon by Simpson's, or other multipliers (as the examples will show), and these products are again multiplied by the various co-sines of the angles of inclination. [It will be noted that the

initial radial line at any angle of heel (*i.e.*, the line forming the water section at any angle of heel) is always the one multiplied by the co-sine, 0°.] After multiplication by the co-sines the products are added together and then divided by 3, as only the third part of the cubes of the ordinates was required. The quotient is multiplied by one-third the angular interval (this, however, according to the multipliers used to affect the products of the cubes of the ordinates); this product is then further multiplied by the longitudinal interval of 9ft. This yields the moment of the wedges uncorrected, and the correction has to be made as heretofore described. After the moment has been corrected it is divided by the displacement expressed in cubic feet, and the quotient is the length of the line E G, Fig. on page 401 (not to be confused with E E_2).

The distance x G can now be readily found, as the distance the centre of gravity is above the centre of buoyancy is known; multiply this distance E F (Fig. 16, page 28) by the sine of the angle of heel, and deduct* the product from the length of the E G; the remainder is the length of the coupling lever, x G: this quantity (the length x G), multiplied by the displacement in tons, expresses the statical stability or righting moment in foot tons. In Sea Belle the centre of buoyancy of 155 tons displacement was found to be 3·4ft. below the load water-line. The meta-centre (calculated from the sum of the products of cubes of the L.W.L. after division by 3) was found to be 4·1ft. above the centre of buoyancy. It has been shown on page 376 that the centre of gravity of Sea Belle is 3·3235ft. below the meta-centre; deduct this from 4·104ft. (height of meta-centre), and the remainder of 0·78ft. is the distance the centre of gravity is above the centre of buoyancy (a little more than 9 inches).

It now only remains to put the different lengths, x G, as so many lines perpendicular to a scale on which is set out the angular intervals at which the stability was calculated the lengths x G will be unequal, and a curve run in through their terminations will make the "curve of stability."

In the "Tables of Stability" it will be noted that for 10° inclination the rule in page 375 is used. Care must be taken to insure that the proper multipliers and divisors to suit the number of angular intervals are used in all cases. [It will be best, to avoid error, to so manage that one of the radial lines as nearly as possible cuts the deck edge.]

The calculation of the "Dynamical Stability," or the amount of work done in heeling the vessel, will now be a comparatively simple operation. The "work" means the force exerted to separate the centre of gravity and centre of buoyancy from their original relative vertical positions. It has been abundantly proved that the centre of buoyancy

* If the C. G. falls below the C. B. the product must be added.

shifts out to leeward upon the inclination of the vessel, and thus puts in operation the balance of forces described as "statical stability." It can be equally proved that the centre of gravity of the vessel is lifted during this inclination; this much can easily be seen by referring to Fig. on page 401, F (the centre of gravity), it will be found, is farther removed from L.W.S., the load water-line of the upright position, than from W.S. 20°, the load water-line of the inclined position; also, if a line were produced through E_2 parallel to W.S. 20°, it would cut the vertical line through F at some point below x. The vertical distance between this point and G = E_2 G will show the extent of the increase that has taken place in the separation of the centre of gravity and centre of buoyancy during the inclination. The distance, multiplied by the weight in tons of the ship, will give the amount of the work done in foot tons. This does not include ordinary keel resistance nor the resistance due to bilge keels,* nor the resistance due to surface friction; but it is on the safe side to conclude that the keel resistance has been included in the calculation. The use of determining the exact value of a vessel's dynamical stability is to know to what extent she will resist any sudden application of wind which would take her beyond what would be her permanent angle of heel if the same force of wind were applied steadily.

The calculation is thus made: The various products for the moment of the wedge inserted in the eighth column are multiplied by the sine of the angle of inclination for which the wedge has been calculated; the products are placed in the last column, and, having been summed, are divided by 3; the quotient is then multiplied by the quantity of the angular interval in circular measure, used in the calculation for statical stability. This product is multiplied by the longitudinal interval, and the moment thus arrived at being "corrected" is divided by the displacement in cubic feet—the quotient is the distance E 2 G (page 401). The quantity E F is then multiplied by the versed sine of the angle of inclination, and subtracted from E 2 G; the remainder multiplied by the displacement in tons represents the work done in foot tons in heeling the vessel to any given angle.

* At the time the inquiry was held into the loss of the Captain someone raised the question as to whether keels and bilge keels would add to stiffness under canvas: it was properly pointed out at this inquiry that, so far as keels or bilge keels of wood are concerned, they tend to decrease statical stability, but on account of the resistance they offer to motion in the water they would check an immediate inclination of the vessel due to a sudden application of wind force by increasing the "amount of work to be done" in heeling; in other words, they would increase the dynamical stability. However, as further pointed out at the inquiry, the lee bilge keel will have a tendency, when the vessel is sailing with a steady wind pressure, to cause an increase of heel beyond that due to the actual pressure on the sails. A vessel when sailing with the wind abeam or forward of the beam makes more or less leeway or moves in a sideways direction; thus, an ardent pressure would be brought upon the upper side of the lee bilge keel, and this pressure would assist in a small degree in heeling the vessel.

FIRST TABLE OF ORDINATES OF WATER SECTIONS.

Water Section at Load Water-line.

Immersed Section.

Nos. of Ordinates.	Lengths of Ordinates.	Simpson's Multipliers.	Products of Ordinates.	Squares of Ordinates.	Simpson's Multipliers.	Products of Squares.	Cubes of Ordinates.	Simpson's Multipliers.	Products of Cubes.
1	0·2	1	0·2	0·0	1	0·0	0·0	1	0·0
2	2·3	4	9·2	5·2	4	20·8	12·0	4	48·0
3	4·8	2	9·6	23·0	2	46·0	110·6	2	221·2
4	7·0	4	28·0	49·0	4	196·0	343·0	4	1372·0
5	8·6	2	17·2	74·0	2	148·0	636·0	2	1272·0
6	9·4	4	37·6	88·3	4	353·2	830·5	4	3322·0
7	9·3	2	18·6	86·4	2	172·8	804·3	2	1608·6
8	8·5	4	34·0	72·2	4	288·8	614·0	4	2456·0
9	6·8	2	13·6	46·2	2	92·4	314·4	2	628·8
10	4·0	4	16·0	16·0	4	64·0	64·0	4	256·0
11	0·3	1	0·3	0·0	1	0·0	0·0	1	0·0
			3)184·3			3)1382·0	3728·8		3)11184·6
			61·4			460·6			3728·2
									2
						Sum of both wedges ...			7456·4

Emersed Section.

The calculations for the emersed section are the same as those for the immersed.

Water Section at 10° Inclination.

Immersed Section.

Nos. of Ordinates.	Lengths of Ordinates.	Simpson's Multipliers.	Products of Ordinates.	Squares of Ordinates.	Simpson's Multipliers.	Products of Squares.	Cubes of Ordinates.	Simpson's Multipliers.	Products of Cubes.
1	0·2	1	0·2	0·0	1	0·0	0·0	1	0·0
2	2·5	4	10·0	6·2	4	24·8	15·6	4	62·4
3	5·2	2	10·4	27·0	2	54·0	140·6	2	281·2
4	7·6	4	30·4	57·7	4	230·8	439·0	4	1756·0
5	9·1	2	18·2	82·8	2	165·6	753·5	2	1507·0
6	9·6	4	38·4	92·1	4	368·4	884·7	4	3538·8
7	9·6	2	19·2	92·1	2	184·2	884·7	2	1769·4
8	9·0	4	36·0	81·0	4	324·0	729·0	4	2916·0
9	7·8	2	15·6	60·8	2	121·6	474·5	2	949·0
10	5·1	4	20·4	26·0	4	104·0	132·6	4	530·4
11	0·3	1	0·3	0·0	1	0·0	0·0	1	0·0
			3)199·1			3)1577·4	4454·2		3)13310·2
			66·3			525·8			4436·7
									3035·1
						Sum of both wedges ...			7471·8

Emersed Section.

Nos. of Ordinates.	Lengths of Ordinates.	Simpson's Multipliers.	Products of Ordinates.	Squares of Ordinates.	Simpson's Multipliers.	Products of Squares.	Cubes of Ordinates.	Simpson's Multipliers.	Products of Cubes.
1	0·2	1	0·2	0·0	1	0·0	0·0	1	0·0
2	2·2	4	8·8	4·8	4	19·2	10·4	4	41·6
3	4·6	2	9·2	21·1	2	42·2	97·3	2	194·6
4	6·6	4	26·4	43·5	4	174·0	287·4	4	1149·6
5	8·1	2	16·2	65·6	2	131·2	531·4	2	1062·8
6	8·9	4	35·6	79·2	4	316·8	705·0	4	2820·0
7	8·8	2	17·6	77·4	2	154·8	681·4	2	1362·8
8	7·8	4	31·2	60·8	4	243·2	474·5	4	1898·0
9	6·0	2	12·0	36·0	2	72·0	216·0	2	432·0
10	3·3	4	13·2	10·9	4	43·6	36·0	4	144·0
11	0·3	1	0·3	0·0	1	0·0	0·0	1	0·0
			3)170·7			3)1197·0			3)9105·4
			56·9			399·0			3035·1

Water Section at 20° Inclination.

Immersed Section.

Nos. of Ordinates.	Lengths of Ordinates.	Simpson's Multipliers.	Products of Ordinates.	Squares of Ordinates.	Simpson's Multipliers.	Products of Squares.	Cubes of Ordinates.	Simpson's Multipliers.	Products of Cubes.
1	0·2	1	0·2	0·0	1	0·0	0·0	1	0·0
2	2·8	4	11·2	7·8	4	31·2	22·0	4	88·0
3	5·9	2	11·8	34·8	2	69·6	205·3	2	410·6
4	8·4	4	33·6	70·5	4	282·0	592·7	4	2370·8
5	9·6	2	19·2	92·1	2	184·2	884·7	2	1769·4
6	10·0	4	40·0	100·0	4	400·0	1000·0	4	4000·0
7	9·9	2	19·8	98·0	2	196·0	970·3	2	1940·6
8	9·5	4	38·0	90·2	4	360·8	857·3	4	3429·2
9	8·6	2	17·2	74·0	2	148·0	636·0	2	1272·0
10	7·0	4	28·0	49·0	4	196·0	343·0	4	1372·0
11	0·3	1	0·3	0·0	1	0·0	0·0	1	0·0
			3)219·3			3)1867·8	5511·3		3)16652·6
			73·1			622·6			5550·8
									2394·8
						Sum of both wedges ...			7945·6

Emersed Section.

Nos. of Ordinates.	Lengths of Ordinates.	Simpson's Multipliers.	Products of Ordinates.	Squares of Ordinates.	Simpson's Multipliers.	Products of Squares.	Cubes of Ordinates.	Simpson's Multipliers.	Products of Cubes.
1	0·2	1	0·2	0·0	1	0·0	0·0	1	0·0
2	2 2	4	8·8	4·8	4	19·2	10·6	4	42·4
3	4·4	2	8·8	19·3	2	38·6	85·2	2	170·4
4	6·2	4	24·8	38·4	4	153·6	238·3	4	953·2
5	7·7	2	15·4	59·3	2	118·6	456·5	2	913·0
6	8·2	4	32·8	67·2	4	268·8	551·3	4	2205·2
7	8·1	2	16·2	65·6	2	131·2	531·4	2	1062·8
8	7·1	4	28·4	50·4	4	201·6	358·0	4	1432·0
9	5·3	2	10·6	28·1	2	56·2	148·8	2	297·6
10	3·0	4	12·0	9·0	4	36·0	27·0	4	108·0
11	0·3	1	0·3	0·0	1	0·0	0·0	1	0·0
			3)158·3			3)1023·8			3)7184·6
			52·7			341·2			2394·8

Water Section at 30° Inclination.

Immersed Section.

Nos. of Ordinates.	Lengths of Ordinates.	Simpson's Multipliers.	Products of Ordinates.	Squares of Ordinates.	Simpson's Multipliers.	Products of Squares.	Cubes of Ordinates.	Simpson's Multipliers.	Products of Cubes.
1	0·2	1	0·2	0·0	1	0·0	0·0	1	0·0
2	3·2	4	12·8	10·2	4	40·8	32·7	4	130·8
3	6·9	2	13·8	47·6	2	95·2	328·5	2	657·0
4	9·3	4	37·2	86·5	4	346·0	804·3	4	3217·2
5	8·6	2	17·2	74·0	2	148·0	636·0	2	1272·0
6	7·7	4	30·8	59·3	4	237·2	456·5	4	1826·0
7	7·4	2	14·8	54·7	2	109·4	405·2	2	810·4
8	7·0	4	28·0	49·0	4	196·0	343·0	4	1372·0
9	6·8	2	13·6	46·8	2	93·6	314·4	2	628·8
10	7·2	4	28·8	51·8	4	207·2	373·2	4	1492·8
11	0·3	1	0·3	0·0	1	0·0	0·0	1	0·0
			3)197·5			3)1473·4			3)11407·0
			65·8			491·1			3802·3
									1947·0
						Sum of both wedges ...			5749·3

Emersed Section.

Nos. of Ordinates.	Lengths of Ordinates.	Simpson's Multipliers.	Products of Ordinates.	Squares of Ordinates.	Simpson's Multipliers.	Products of Squares.	Cubes of Ordinates.	Simpson's Multipliers.	Products of Cubes.
1	0·2	1	0·2	0·0	1	0·0	0·0	1	0·0
2	2·3	4	9·2	5·3	4	21·2	12·1	4	48·4
3	4·4	2	8·8	19·3	2	38·6	85·1	2	170·2
4	6·0	4	24·0	36·0	4	144·0	216·0	4	864·0
5	7·2	2	14·4	51·8	2	103·6	373·2	2	746·4
6	7·6	4	30·4	57·7	4	230·8	439·0	4	1756·0
7	7·4	2	14·8	54·7	2	109·4	405·2	2	810·4
8	6·5	4	26·0	42·2	4	168·8	274·6	4	1098·4
9	5·0	2	10·0	25·0	2	50·0	125·0	2	250·0
10	2·9	4	11·6	8·4	4	33·6	24·3	4	97·2
11	0·3	1	0·3	0·0	1	0·0	0·0	1	0·0
			3)149·7			3)900·0			3)5841·0
			49·9			300·0			1947·0

Water Section at 40° Inclination.

Immersed Section.

Nos. of Ordinates.	Lengths of Ordinates.	Simpson's Multipliers.	Products of Ordinates.	Squares of Ordinates.	Simpson's Multipliers.	Products of Squares.	Cubes of Ordinates.	Simpson's Multipliers.	Products of Cubes.
1	0·2	1	0·2	0·0	1	0·0	0·0	1	0·0
2	4·0	4	16·0	16·0	4	64·0	64·0	4	256·0
3	8·4	2	16·8	70·5	2	141·0	592·7	2	1185·4
4	7·6	4	30·4	57·7	4	230·0	439·0	4	1756·0
5	6·9	2	13·8	47·6	2	95·2	328·5	2	657·0
6	6·1	4	24·4	37·2	4	148·3	227·0	4	908·0
7	5·8	2	11·6	33·6	2	67·2	195·1	2	390·2
8	5·4	4	21·6	29·1	4	116·4	157·4	4	629·6
9	5·2	2	10·4	27·0	2	54·0	140·6	2	281·2
10	5·6	4	22·4	31·3	4	125·2	175·6	4	702·4
11	6·4	1	6·4	41·0	1	41·0	262·1	1	262·1
			3)174·0			3)1083·6			3)7027·9
			58·0			361·2			2342·6
									1708·2
						Sum of both wedges ...			4050·8

Emersed Section.

Nos. of Ordinates.	Lengths of Ordinates.	Simpson's Multipliers.	Products of Ordinates.	Squares of Ordinates.	Simpson's Multipliers.	Products of Squares.	Cubes of Ordinates.	Simpson's Multipliers.	Products of Cubes.
1	0·2	1	0·2	0·0	1	0·0	0·0	1	0·0
2	2·4	4	9·6	5·7	4	22·8	13·8	4	55·2
3	4·5	2	9·0	20·2	2	40·4	91·1	2	182·2
4	6·0	4	24·0	36·0	4	144·0	216·0	4	864·0
5	6·9	2	13·8	47·6	2	95·2	328·5	2	657·0
6	7·2	4	28·8	51·8	4	207·2	373·2	4	1492·8
7	6·9	2	13·8	47·6	2	95·2	328·5	2	657·0
8	6·1	4	24·4	37·2	4	148·8	227·0	4	908·0
9	4·8	2	9·6	23·0	2	46·0	110·5	2	221·0
10	2·8	4	11·2	7·8	4	31·2	21·9	4	87·6
11	0·3	1	0·3	0·0	1	0·0	0·0	1	0·0
			3)144·7			3)830·8			3124·8
			48·2			276·9			1708·2

Water Section at 50° Inclination.

Immersed Section.

Nos. of Ordinates.	Lengths of Ordinates.	Simpson's Multipliers.	Products of Ordinates.	Squares of Ordinates.	Simpson's Multipliers.	Products of Squares.	Cubes of Ordinates.	Simpson's Multipliers.	Products of Cubes.
1	0·2	1	0·2	0·0	1	0·0	0 0	1	0·0
2	5·5	4	22·0	30·2	4	120·8	166 3	4	665·2
3	7·6	2	15·2	57·7	2	115·4	439·0	2	878·0
4	6·5	4	26·0	42·2	4	168·8	274·6	4	1098·4
5	5·8	2	11·6	33·6	2	67·2	195·1	2	390·2
6	5·2	4	20·8	27·0	4	108·0	140·6	4	562·4
7	4·9	2	9·8	24·0	2	48·0	117·6	2	235·2
8	4·6	4	18·4	21·1	4	84·4	97 3	4	389·2
9	4·5	2	9·0	20·2	2	40·4	91·1	2	182·2
10	4·8	4	19·2	23·0	4	92·0	110·6	4	442·4
11	5·4	1	5·4	29·1	1	29·9	157·4	1	157·4
			3)157·6			3)874·1			3)5000·6
			52·5			291·3			1666·9
									1668·7
						Sum of both wedges ...			3335·6

Emersed Section.

Nos. of Ordinates.	Lengths of Ordinates.	Simpson's Multipliers.	Products of Ordinates.	Squares of Ordinates.	Simpson's Multipliers.	Products of Squares.	Cubes of Ordinates.	Simpson's Multipliers.	Products of Cubes.
1	0·2	1	0·2	0·0	1	0·0	0·0	1	0·0
2	2·7	4	10·8	7·3	4	29·2	19·6	4	78·4
3	4·8	2	9·6	23·0	2	46·0	110·6	2	221·2
4	6·1	4	24·4	37·2	4	148·8	227·0	4	908·0
5	6·8	2	13·6	46·2	2	92·4	314·4	2	628·8
6	7·0	4	28·0	49·0	4	196·0	343·0	4	1372·0
7	6·7	2	13·4	44·9	2	89·8	300·7	2	601·4
8	6·0	4	24·0	36·0	4	144·0	216·0	4	864·0
9	4·9	2	9·8	24·0	2	48·0	117·6	2	235·2
10	2·9	4	11·6	8·4	4	33·6	24·3	4	97·2
11	0·3	1	0·3	0·0	1	0·0	0·0	1	0·0
			3)145·7			3)827·8			3)5006·2
			48·5			275·9			1668·7

Water Section at 60° Inclination.

Immersed Section.

Nos. of Ordinates.	Lengths of Ordinates.	Simpson's Multipliers.	Products of Ordinates.	Squares of Ordinates.	Simpson's Multipliers.	Products of Squares.	Cubes of Ordinates.	Simpson's Multipliers.	Products of Cubes.
1	0·2	1	0·2	0·0	1	0·0	0·0	1	0·0
2	7·0	4	28·0	49·0	4	196·0	343·0	4	1372·0
3	6·3	2	12·6	39·7	2	79·4	250·0	2	500·0
4	5·8	4	23·2	33·6	4	134·4	195·1	4	780·4
5	5·2	2	10·4	27·0	2	54·0	140·6	2	281·2
6	4·7	4	18·8	22·1	4	88·4	103·8	4	415·2
7	4·3	2	8·6	18·5	2	37·0	79·5	2	159·0
8	4·1	4	16·4	16·8	4	67·2	68·9	4	275·6
9	3·9	2	7·8	15·2	2	30·4	59·3	2	118·6
10	4·2	4	16·8	17·6	4	70·4	74·0	4	296·0
11	5·8	1	5·8	33·6	1	35·6	195·0	1	195·0
			3)148·6			3)790·8			3)4393·0
			49·5			236·6			1464·3
									1852·7
						Sum of both wedges ...			3317·0

Emersed Section.

Nos. of Ordinates.	Lengths of Ordinates.	Simpson's Multipliers.	Products of Ordinates.	Squares of Ordinates.	Simpson's Multipliers.	Products of Squares.	Cubes of Ordinates.	Simpson's Multipliers.	Products of Cubes.
1	0·2	1	0·2	0·0	1	0·0	0·0	1	0·0
2	3·1	4	12·4	9·6	4	38·4	29·8	4	119·2
3	5·2	2	10·4	27·0	2	54·0	140·6	2	281·2
4	6·4	4	25·6	41·0	4	164·0	262·1	4	1048·4
5	7·0	2	14·0	49·0	2	98·0	343·0	2	686·0
6	7·1	4	28·4	50·4	4	201·6	358·0	4	1432·0
7	6·8	2	13·6	46·2	2	92·4	314·4	2	628·8
8	6·2	4	24·8	38·4	4	153·6	238·3	4	953·2
9	5·1	2	10·2	26·0	2	52·0	132·6	2	265·2
10	3·3	4	13·2	10·9	4	43·6	36·0	4	144·0
11	0·3	1	0·3	0·0	1	0·0	0·0	1	0·0
			3)153·1			3)897·6			3)5558·0
			51·0			299·2			1852·7

Water Section at 70° Inclination.

Immersed Section.

Nos. of Ordinates.	Lengths of Ordinates.	Simpson's Multipliers.	Products of Ordinates.	Squares of Ordinates.	Simpson's Multipliers.	Products of Squares.	Cubes of Ordinates.	Simpson's Multipliers.	Products of Cubes.
1	0·2	1	0·2	0·0	1	0·0	0·0	1	0·0
2	6·5	4	26·0	42·2	4	168·8	274·6	4	1098·4
3	5·9	2	11·8	34·8	2	69·6	205·3	2	410·6
4	5·4	4	21·6	29·1	4	116·4	157·4	4	629·6
5	4·9	2	9·8	24·0	2	48·0	117·6	2	235·2
6	4·3	4	17·2	18·5	4	74·0	79·5	4	318·0
7	4·3	2	8·6	18·5	2	37·0	79·5	2	159·0
8	4·1	4	16·4	16·8	4	67·2	68·9	4	275·6
9	3·9	2	7·8	15·2	2	30·4	59·3	2	118·6
10	4·2	4	16·8	17·6	4	70·4	74·0	4	296·0
11	4·8	1	4·8	23·0	1	23·0	110·6	1	110·6
			3)141·0			3)704·8			3)3651·6
			47·0			234·9			1217·2
									2369·3
						Sum of both wedges ...			3586·5

Emersed Section.

Nos. of Ordinates.	Lengths of Ordinates.	Simpson's Multipliers.	Products of Ordinates.	Squares of Ordinates.	Simpson's Multipliers.	Products of Squares.	Cubes of Ordinates.	Simpson's Multipliers.	Products of Cubes.
1	0·2	1	0·2	0·0	1	0·0	0·0	1	0·0
2	3·9	4	15·6	15·2	4	60·8	59·3	4	237·2
3	6·0	2	12·0	36·0	2	72·0	216·0	2	432·0
4	7·0	4	28·0	49·0	4	196·0	343·0	4	1372·0
5	7·4	2	14·8	54·7	2	109·4	405·2	2	810·4
6	7·5	4	30·0	56·2	4	224·8	421·8	4	1687·2
7	7·3	2	14·6	53·3	2	106·6	389·0	2	778·0
8	6·7	4	26·8	44·9	4	179·6	300·7	4	1202·8
9	5·6	2	11·2	31·3	2	62·6	175·6	2	351·2
10	3·9	4	15·6	15·2	4	60·8	59·3	4	237·2
11	0·3	1	0·3	0·0	1	0·0	0·0	1	0·0
			3)169·1			3)1072·6			3)7108·0
			56·3			357·5			2369·3

Water Section at 80° Inclination.

Immersed Section.

Nos. of Ordinates.	Lengths of Ordinates.	Simpson's Multipliers.	Products of Ordinates.	Squares of Ordinates.	Simpson's Multipliers.	Products of Squares.	Cubes of Ordinates.	Simpson's Multipliers.	Products of Cubes.
1	0·2	1	0·2	0·0	1	0·0	0·0	1	0·0
2	6·2	4	24·8	38·4	4	153·6	238·3	4	953·2
3	5·6	2	11·2	31·3	2	62·6	175·6	2	351·2
4	5·2	4	20·8	27·0	4	108·0	140·6	4	562·4
5	4·6	2	9·2	21·1	2	42·2	97·3	2	194·6
6	4·2	4	16·8	17·6	4	70·4	74·0	4	296·0
7	3·8	2	7·6	14·4	2	28·8	54·8	2	109·6
8	3·6	4	14·4	13·0	4	52·0	46·6	4	186·4
9	3·5	2	7·0	12·2	2	24·4	42·8	2	85·6
10	3·7	4	14·8	13·7	4	54·8	50·6	4	202·4
11	4·2	1	4·2	17·6	1	17·6	74·0	1	74·0
			3)131·0			3)614·4			3)3015·4
			43·6			204·8			1005·1
									3940·7
						Sum of both wedges ...			4945·8

Emersed Section.

Nos. of Ordinates.	Lengths of Ordinates.	Simpson's Multipliers.	Products of Ordinates.	Squares of Ordinates.	Simpson's Multipliers.	Products of Squares.	Cubes of Ordinates.	Simpson's Multipliers.	Products of Cubes.
1	0·2	1	0·2	0·0	1	0·0	0·0	1	0·0
2	5·3	4	21·2	28·1	4	112·4	148·8	4	595·2
3	7·4	2	14·8	54·7	2	109·4	405·2	2	810·4
4	8·2	4	32·8	67·2	4	268·8	551·3	4	2205·2
5	8·4	2	16·8	70·5	2	141·0	592·7	2	1185·4
6	8·6	4	34·4	74·0	4	296·0	636·0	4	2544·0
7	8·4	2	16·8	70·5	2	141·0	592·7	2	1185·4
8	8·0	4	32·0	64·0	4	256·0	512·0	4	2048·0
9	7·0	2	14·0	49·0	2	98·0	343·0	2	686·0
10	5·2	4	20·8	27·0	4	108·0	140·6	4	562·4
11	0·3	1	0·3	0·0	1	0·0	0·0	1	0·0
			3)204·1			3)1530·6			3)11822·0
			68·0			510·2			3940·7

Water Section at 90° Inclination.

Immersed Section.

Nos. of Ordinates.	Lengths of Ordinates.	Simpson's Multipliers.	Products of Ordinates.	Squares of Ordinates.	Simpson's Multipliers.	Products of Squares.	Cubes of Ordinates.	Simpson's Multipliers.	Products of Cubes.
1	0·0	1	0·0	0·0	1	0·0	0·0	1	0·0
2	6·2	4	24·8	38·4	4	153·6	238·3	4	953·2
3	5·6	2	11·2	31·6	2	63·2	175·6	2	351·2
4	5·1	4	20·4	26·0	4	104·0	132·6	4	530·4
5	4·6	2	9·2	21·1	2	42·2	97·3	2	194·6
6	4·1	4	16·4	16·8	4	67·2	68·9	4	275·6
7	3·7	2	7·4	13·7	2	27·4	50·6	2	101·2
8	3·5	4	14·0	12·5	4	50·0	42·8	4	171·2
9	3·4	2	6·8	11·5	2	23·0	39·3	2	78·6
10	3·6	4	14·4	13·0	4	52·0	46·6	4	186·4
11	4·1	1	4·1	16·8	1	16·8	68·9	1	68·9
			3)128·7			3)599·4			3)2911·3
			42·9			199·8			970·4
									13784·5
						Sum of both wedges ...			14754·9

Emersed Section.

Nos. of Ordinates.	Lengths of Ordinates.	Simpson's Multipliers.	Products of Ordinates.	Squares of Ordinates.	Simpson's Multipliers.	Products of Squares.	Cubes of Ordinates.	Simpson's Multipliers.	Products of Cubes.
1	0·0	1	0·0	0·0	1	0·0	0·0	1	0·0
2	8·5	4	34·0	72·2	4	288·8	614·0	4	2456·0
3	10·3	2	20·6	106·0	2	212·0	1092·7	2	2185·4
4	11·0	4	44·0	121·0	4	484·0	1331·0	4	5324·0
5	11·5	2	23·0	132·2	2	264·4	1520·8	2	3041·6
6	11·8	4	47·2	139·2	4	556·8	1643·0	4	6572·0
7	12·0	2	24·0	144·0	2	288·0	1728·0	2	3456·0
8	12·2	4	48·8	148·8	4	595·2	1815·8	4	7263·2
9	12·3	2	24·6	151·3	2	302·6	1860·8	2	3721·6
10	12·0	4	48·0	144·0	4	576·0	1728·0	4	6912·0
11	7·5	1	7·5	56·2	1	56·2	421·8	1	421·8
			3)321·7			3)3624·0			3)41353·6
			107·2			1208·0			13784·5

TABLES OF STABILITY.

Stability at 10° Inclination.

Immersed Wedge.					Both Wedges.			Statical Stability.		Dynamical Stability.	
Degrees of Inclination.	Products of Ordinates.	Products of Squares of Ordinates.	Multipliers.	Products.	Sums of Products of Cubes of Ordinates of both Wedges.	Multipliers.	Products for Moments of Wedges.	Co-sines of Angles of Inclination.	Products for Moments of Wedges × co-sine θ.	Sines of Angle of Inclination.	Products for Moments of Wedges × sine θ.
0°	...	460·6	5	2303·0	7456·4	5	37282·0	·9848	36715·3	·1736	6472
10°	66·3	525·8	8	4206·4	7471·7	8	59773·6	1·0000	59773·6	·0000	·0000
				6509·4					96488·9		6472
20°	...	622·6	1	− 622·6	7945·6	1	7945·6	·9848	− 7824·8	·1736	1379

Immersed Wedge (continued).

5886·8
5153·8
2)733·0
366·5
× $\frac{1}{12}$ angular interval ... ·0145
·18325
1·4660
3·665
5·31425
× longitudinal interval... 9
Excess in volume of immersion .... = 47·82825

Statical Stability (continued).

3)88664·1
29554·7
× $\frac{1}{12}$ angular interval .... ·0145
14·77735
118·2188
295·547
428·54315
× longitudinal interval 9
Moment of wedges uncorrected = 3856·88835
Correction for layer .... − 24·58359
÷ displacement .... 5425)3832·30476(·7062

E F = distance C. G. is above C. B.

E x = E F × sin. θ = ·78 × ·1736 .... ·1736
·78
·013888
·12152
·135408

E G .... = ·706200
− E x .... ·135408
x G .... = ·570792
× displacement..................... 155 tons
2·853960
28·53960
57·0792
Righting moment in foot tons = 88·472760

Dynamical Stability (continued).

3)5093
1698
·0145
·8490
6·792
16·98
24·6210
9
221·5890
− 1·0038
5425)220·5852(·0466

E F × vers. θ
= ·78 × ·0151
·78
1208
1057
·011778

E$_2$ G = ·040660
·011778
·028882 × Displacement.
155
·144410
1·44410
2·8882
4·476710*

* 4·4767 = work in foot tons.

EMERSED WEDGE.

0°	...	460·6	5	2303·0
10°	56·9	399·0	8	3192·0
				5495·0
20°	...	341·2	1	− 341·2
				5153·8

AREA OF INCLINED WATER SECTION.

Immersed section .... 66·3
Emersed section .... 56·9
123·2
× longitudinal interval... 9
Area ....= 1108·8 sq. ft.

CENTRE OF GRAVITY OF THE INCLINED WATER SECTION.

Products of Squares.
Immersed .... 525·8
Emersed .... 399·0
2)126·8
63·4
× longitudinal interval 9
÷ area .... 1108·8)570·60(·514*

* 0·514ft. = the distance the centre of gravity of the inclined water section is on the immersed side of the section.

LAYER.

Excess in immersions ÷ area of inclined section =
1108·8)47·828(·043
44·352
3·4760
3·3264
·1496
Thickness of layer = ·043ft.

CORRECTION FOR STATICAL STABILITY.

Volume of layer = 47·828 × distance C.G. is towards the immersed side 0·514ft. = 24·583592 = correction.

CORRECTION FOR DYNAMICAL STABILITY.

Volume of layer 47·8 × ½ thickness of layer ·043ft. = 47·8 × ·021 = 1·0038 = correction.

STABILITY AT 20° INCLINATION.

IMMERSED WEDGE.

Degrees of Inclination.	Products of Ordinates.	Products of Squares of Ordinates.	Multipliers.	Products.
0°	...	460·6	1	460·6
10°	...	525·8	4	2103·2
20°	73·1	622·6	1	622 6

Immersed wedge .... 3186·4
Emersed wedge .... 2397·8

2)788·6

394·3
× ⅓ angular interval ...... ·058

3·1544
19·715

22·8694
× longitudinal interval... 9

Excess in volume of immersion .... = 205·8246

BOTH WEDGES.

Sums of Products of Cubes of Ordinates of both Wedges.	Multipliers.	Products for Moments of Wedges.	Statical Stability: Co-sines of Angles of Inclination.	Statical Stability: Products for Moments of Wedges × co-sine θ.	Dynamical Stability: Sines of Angle of Inclination.	Dynamical Stability: Products for Moments of Wedges × sine θ.
7456·4	1	7456·4	·940	7009·0	·3420	2550
7471·7	4	29886·8	·985	29438·5	·1736	5188
7945·6	1	7945·6	1·000	7945·6	·0000	0000

Statical Stability:

3)44393·1

14797·7
× ⅓ angular interval............... ·058

118·8816
739·885

858·2666
× longitudinal interval 9

Moment of wedges uncorrected = 7724·3994
Correction for layer .... − 205·82

+ displacement .... 5425)7518·5794(1·3859

E x = E F × sin. θ = ·78 × ·342......... .. ·342
·78

2736
·2394

·26676

E G .... = 1·38590
− E x .... ·26676

x G .... = 1·11914
155 tons.

5·59570
55·9570
111·914

Righting moment in foot tons = 173·46370

Dynamical Stability:

3)7738

2579
·058

20·632
128·95

149·582
9

1346·238
— 18·522

5425)1327·716(24

E F × vers. θ
= ·78 × 0607
78

4856
4249

047346

E_2 G = ·2400
·0473

·1927
155

·9635
9·635
19·27

29·8685*

* 29·8685 = work in foot tons.

EMERSED WEDGE.

0°	...	460·6	1	460·6
10°	...	399·0	4	1596·0
20°	52·7	341·2	1	341·2
				2397·8

AREA OF INCLINED WATER SECTION.

Immersed section .... 73·1
Emersed section .... 52·7

125·8
× longitudinal interval... 9

Area ....= 1132·2 sq. ft.

CENTRE OF GRAVITY OF THE INCLINED WATER SECTION.

Products of Squares.
Immersed .... 622·6
Emersed .... 341·2

281·4
× longitudinal interval 9

2)2532·6

÷ area .... 1132·2)1266·3(1·1*

* 1·1ft. = the distance the centre of gravity of the inclined water plane is on the immersed side of the section.

LAYER.

Excess in immersions ÷ area of inclined section =

1132·2)205·82(0·18
113·22

92·600
90·576

2·024

Thickness of layer = 0·18ft.

CORRECTION FOR STATICAL STABILITY.

Volume of layer × distance C.G. is towards the immersed side 205·82 × 1 = 205·82 = correction.

CORRECTION FOR DYNAMICAL STABILITY.

Volume of layer 205·8 × ½ thickness ·18ft. = 205·8 × ·09 = 18·522 = correction.

Stability at 30° Inclination.

Immersed Wedge.					Both Wedges.			Statical Stability.		Dynamical Stability.	
Degrees of Inclination.	Products of Ordinates.	Products of Squares of Ordinates.	Multipliers.	Products.	Sums of Products of Cubes of Ordinates of both Wedges.	Multipliers.	Products for Moments of Wedges.	Co-sines of Angles of Inclination.	Products for Moments of Wedges × co-sine θ.	Sines of Angle of Inclination.	Products for Moments of Wedges × sine θ.
0°	...	460·6	1	460·6	7456·4	1	7456·4	·866	6457·2	·5000	3728·2
10°	...	525·8	3	1577·4	7471·8	3	22415·4	·940	21070·4	·3420	7666·0
20°	...	622·6	3	1867·8	7945·6	3	23836·8	·985	23479·2	·1736	4138·0
30°	65·8	491·1	1	491·1	5749·3	1	5749·3	1·000	5749·3	·0000	0000·0

Immersed Wedge.

4396·9
2981·2
2)1415·7
707·8
× ⅓ angular interval ...... ·0654
·28312
3·5390
42·468
46·29012
× longitudinal interval... 9
Excess in volume of immersion .... = 416·61106

Statical Stability.

3)56756·1
18918·7
× ⅓ angular interval.............. ·0654
7·56748
94·5985
1135·122
1237·28298
× longitudinal interval 9
Moment of wedges uncorrected =11135·54682
Correction for layer .... − 344·11160
÷ displacement .... 5425)10791·43522(1·989

E x = E F × sin. θ .... = ·78 × ·5
·5
·390

E G .... = 1·989
− E x .... ·390
x G .... = 1·599
155 tons.
7·995
79·95
159·9
Righting moment in foot tons... = 247·845

Dynamical Stability.

3)15532·2
5177·4
·0654
2·07096
25·8870
310·654
338·60196
9
3047·41764
− 83·32
5425)2964·09764(·5

E F × vers. θ
= ·78 × ·134
·78
·1072
·938
·10452

E_2 G = ·5
·1
·4 × 155 tons.
·4
62·0*

* 62·0 = work done in foot tons.

Emersed Wedge.

0°	..	460·6	1	460·6
10°	...	399·0	3	1197·0
20°	...	341·2	3	1023·6
30°	49·9	300·0	1	300·0
				2981·2

AREA OF INCLINED WATER SECTION.

Immersed section .... 65·8
Emersed section .... 49·9
115·7
× longitudinal interval... 9
Area ....= 1041·3 sq. ft.

CENTRE OF GRAVITY OF THE INCLINED WATER SECTION.

Products of Squares.
Immersed .... 491·1
Emersed .... 300·0
2)191·1
95·5
× longitudinal interval 9
÷ area............... 1041·3)859·50(·826*

* ·826ft. = the distance the centre of gravity of the inclined water plane is on the immersed side of the section.

LAYER.

Excess in immersions ÷ area of inclined section =

1041·3)416·60(·4
416·52
8

Thickness of layer = 0·4ft.

CORRECTION FOR STATICAL STABILITY.

Volume of layer × distance C.G. is towards the immersed side = 416·6 × ·826 = 344·1116 = correction.

CORRECTION FOR DYNAMICAL STABILITY.

Volume of layer 416·6 × ½ thickness of layer 0·4ft. = 416·6 × 0·2 = 83·32 = correction.

Stability at 40° Inclination.

Immersed Wedge.					Both Wedges.			Statical Stability.		Dynamical Stability.	
Degrees of Inclination.	Products of Ordinates.	Products of Squares of Ordinates.	Multipliers.	Products.	Sums of Products of Cubes of Ordinates of both Wedges.	Multipliers.	Products for Moments of Wedges.	Co-sines of Angles of Inclination.	Products for Moments of Wedges × co-sine θ.	Sines of Angle of Inclination.	Products for Moments of Wedges × sine θ.
0°	...	460·6	1	460·6	7456·4	1	7456·4	·766	5711·6	·6428	4793·0
10°	...	525·8	4	2103·2	7471·7	4	29886·8	·866	25881·9	·5000	14943·4
20°	...	622·6	2	1245·2	7945·6	2	15891·2	·940	14937·7	·3420	5434·7
30°	...	491·1	4	1964·4	5749·3	4	22997·2	·985	22652·2	·1736	3992·3
40°	58·0	361·2	1	361·2	4055·8	1	4050·8	1·000	4054·8	·0000	0000·0

Immersed wedge.

6184·6
4215·9

2)1918·7

959·3
× ⅓ angular interval ...... ·058

7·6744
47·965

55·6394
× longitudinal interval... 9

Excess in volume of immersions .... = 500·7546

Statical stability.

3)73234·2

24411·4
× ⅓ angular interval.............. ·058

195·2912
1220·570

1415·8612
× longitudinal interval 9

Moment of wedges uncorrected = 12742·7508
Correction for layer .... − 190·2850

+ displacement .... 5425)12552·4758(2·3

E x = E F × sin. θ = ·78 × ·643 ... = ·643
·78

5144
4501

·50154

E G .... = 2·3
− E x .... ·5

x G .... = 1·8
155 tons.

9·0
90
18

Righting moment in foot tons = 279·0

Dynamical stability.

3)29163·4

9721·1
·058

77·7688
486·055

563·8238
9

5074·4142
− 130·1820

5425)4944·2322(·9
4882·5

61·7

E F × vers. θ
= ·78 × ·234
·78

1872
·1638

·18252

E_2 G = ·90
·18

·72 × 155 tons
·72

3·10
108·5

111·60*

* 111·60 = work done in foot tons.

Emersed Wedge.

0°	...	460·6	1	460·6
10°	...	399·0	4	1596·0
20°	...	341·2	2	682·4
30°	...	300·0	4	1200·0
40°	48·2	276·9	1	276·9
				4215·9

Area of Inclined Water Section.

	Products of Ordinates.
Immersed section	58·0
Emersed section	48·2
	106·2
× longitudinal interval...	9
Area	= 955·8 sq. ft.

Centre of Gravity of the Inclined Water Section.

	Products of Squares.
Immersed	361·2
Emersed	276·9
	2)84·3
	42·1
× longitudinal interval	9
÷ area...............	955·8)378·90(0·3849*

* 0·3849ft. = the distance the centre of gravity of the inclined water plane is on the immersed side of the section.

Layer.

Excess in immersions ÷ area of inclined section =

955·8)500·75(·523

Thickness of layer = 0·523ft.

Correction for Statical Stability.

Volume of layer × distance C.G. is towards the immersed side = 500·75 × ·38 = 190·2850 = correction.

Correction for Dynamical Stability.

500·7 Volume.
·26 = ½ thickness of layer.

30·042
100·14

130·182 = correction.

Stability at 50° Inclination.

IMMERSED WEDGE.					BOTH WEDGES.			Statical Stability.		Dynamical Stability.	
Degrees of Inclination.	Products of Ordinates.	Products of Squares of Ordinates.	Multipliers.	Products.	Sums of Products of Cubes of Ordinates of both Wedges.	Multipliers.	Products for Moments of Wedges.	Co-sines of Angles of Inclination.	Products for Moments of Wedges × co-sine θ.	Sines of Angle of Inclination.	Products for Moments of Wedges × sine θ.
0°	...	460·6	½	230·3	7456·4	½	3728·2	·642	2393·5	·7660	2855·8
10°	...	525·8	1	525·8	7471·7	1	7471·7	·766	5723·3	·6428	4802·8
20°	...	622·6	1	622·6	7945·6	1	7945·6	·866	6880·8	·5000	3972·8
30°	...	491·1	1	491·1	5749·3	1	5749·3	·940	5404·3	·3420	1966·2
40°	...	361·2	1	361·2	4050·8	1	4050·8	·985	3990·0	·1736	703·2
50°	52·5	291·3	½	145·6	3335·6	½	1667·8	1·000	1667·8	·0000	000·0

Immersed wedge:

2376·6
−1685·3
2)691·3
345·6
× angular interval......... ·174
1·3824
24·192
34·56
60·1344
× longitudinal interval... 9
Excess in volume of immersion. .... = 541·2096

Both wedges — Statical stability:

3)26059·7
8686·5
× angular interval .... ·174
34·7460
608·055
868·65
1511·4510
× longitudinal interval 9
13603·059
Correction for layer .... − 41·131
+ displacement .... 5425)13561·928(2·499

E x = E F × sin. θ = ·78 × ·766 ... = ·766
·78
6128
5362
·59748

E G .... = 2·499
− E x .... ·597
x G .... = 1·902
155 tons.
9·510
95·10
190·2
Righting moment in foot tons = 294·810

Dynamical stability:

3)14300·8
4766·9
·174
19·0676
333·683
476·69
829·4406
9
7464·9654
— 162·8000
5425)7302·6654(1·34
E F × vers. θ
= ·78 × ·357
·78
2856
·2499
·27846
E_2 G ...... = 1·340
·278
1·062
155
5·310
51·10
106·2
164·610*

* 164·610 = work done in foot tons.

EMERSED WEDGE.

0°	...	460·6	½	230·3
10°	...	399·0	1	399·0
20°	...	341·2	1	341·2
30°	...	300·0	1	300·0
40°	...	276·9	1	276·9
50°	48·5	275·9	½	137·9
				1685·3

AREA OF INCLINED WATER SECTION.

Products of Ordinates.
Immersed section .... 52·5
Emersed section .... 48·5
101·0
× longitudinal interval... 9
Area .... = 909·0 sq. ft.

CENTRE OF GRAVITY OF THE INCLINED WATER SECTION.

Products of Squares.
Immersed .... 291·3
Emersed .... 275·9
2)15·4
7·7
× longitudinal interval 9
+ area .... 909)69·30(·076*

* 0·076ft. = the distance the centre of gravity of the inclined water plane is on the immersed side of the section.

LAYER.

Excess in immersions + area of inclined section =

909)541·2(·595

Thickness of layer = 0·595ft.

CORRECTION FOR STATICAL STABILITY.

Volume of layer × distance C.G. is towards the immersed side = 541·2 × ·076 = 41·1312 = correction.

CORRECTION FOR DYNAMICAL STABILITY.

541 volume of layer.
.3 = ½ thickness of layer.
162·3 = correction.

Stability at 60° Inclination.

Immersed Wedge. — Both Wedges (Statical Stability; Dynamical Stability).

Degrees of Inclination.	Products of Ordinates.	Products of Squares of Ordinates.	Multipliers.	Products.	Sums of Products of Cubes of Ordinates of both Wedges.	Multipliers.	Products for Moments of Wedges.	Statical Stability. Co-sines of Angles of Inclination.	Products for Moments of Wedges × co-sine θ.	Dynamical Stability. Sines of Angle of Inclination.	Products for Moments of Wedges × sine θ.
0°	...	460·6	1	460·6	7456·4	1	7456·4	·500	3728·2	·8660	6457·2
10°	...	525·8	4	2103·2	7471·7	4	29886·8	·642	19187·3	·7660	22898·2
20°	...	622·6	2	1245·2	7945·6	2	15891·2	·766	12172·6	·6428	10214·8
30°	...	491·1	4	1964·4	5749·3	4	22997·2	·866	19915·5	·5000	11498·6
40°	...	361·2	2	722·4	4050·8	2	8101·6	·940	7615·5	·3420	2770·7
50°	...	291·3	4	1165·2	3335·6	4	13342·4	·985	13142·2	·1736	2316·2
60°	49·5	263·6	1	263·6	3317·0	1	3317·0	1·000	3317·0	·0000	0000·0

Immersed wedge:

7924·6
−5895·6
2)2029·0
1014·5
× ⅓ angular interval ...... ·058
8·1160
50·725
58·8419
× longitudinal interval... 9
Excess in volume of immersion . ... = 529·5690

Statical stability:

3)79078·3
26359·4
× ⅓ angular interval.............. ·058
210·8752
1317·970
1528·8452
× longitudinal interval 9
Moment of wedges uncorrected = 13759·6068
Correction for layer .... + 93·7215
+ displacement .... 5425)13853·3283(2·55

E x = E F × sin. θ = ·78 × ·866 ...... ·866
·78
6928
6062
·67548

E G .... = 2·55
− E x .... ·67
x G .... = 1·88
155 tons.
9·40
94·0
188
Righting moment in foot tons = 291·40

Dynamical stability:

3)56150·7
18716·9
·058
149·7352
935·845
1085·5802
9
9770·2218
— 153·41
5425)9616·8118(1·77

E F × vers. θ ·78
= ·78 × ·5
·390

E₂G = 1·77
·39
1·38 × 155 tons.
155
6·90
69·0
138
213·90*

* 213·90 = work done in foot tons.

Emersed Wedge.

0°	...	460·6	1	460·6
10°	...	399·0	4	1596·0
20°	...	341·2	2	682·4
30°	...	300·0	4	1200·0
40°	...	276·9	2	553·8
50°	...	275·9	4	1103·6
60°	51·0	299·2	1	299·2
				5895·6

Area of Inclined Water Section.

Products of Ordinates.
Immersed section .... 49·5
Emersed section .... 51·0
100·5
× longitudinal interval... 9
Area .... = 904·5 sq. ft.

Centre of Gravity of the Inclined Water Section.

Products of Squares.
Immersed .... 299·2
Emersed .... 263·6
2)35·6
17·8
× longitudinal interval 9
÷ area............... 904·5)160·20(·177*

* 0·177ft. = the distance the centre of gravity of the inclined water plane is on the immersed side of the section.

Layer.

Excess in immersions ÷ area of inclined section =

904·5)529·57(·585

Thickness of layer = 0·585ft.

Correction for Statical Stability.

Volume of layer × distance C.G. is towards the immersed side = 529·5 × ·177 = 93·7215 = correction.

Correction for Dynamical Stability.

529 volume of layer.
·29 = ½ thickness of layer.
47·61
105·8
153·41 = correction.

Stability at 70° Inclination.

Immersed Wedge.					Both Wedges.			Statical Stability.		Dynamical Stability.	
Degrees of Inclination.	Products of Ordinates.	Products of Squares of Ordinates.	Multipliers.	Products.	Sums of Products of Cubes of Ordinates of both Wedges.	Multipliers.	Products for Moments of Wedges.	Co-sines of Angles of Inclination.	Products for Moments of Wedges × co-sine θ.	Sines of Angle of Inclination.	Products for Moments of Wedges × sine θ.
0°	...	460·6	½	230·3	7456·4	½	3728·2	.342	1275·0	·9397	3503·3
10°	...	525·8	1	525·8	7471·7	1	7471·7	·500	3735·8	·8660	6470·5
20°	...	622·6	1	622·6	7945·6	1	7945·6	·642	5101·0	·7660	6086·3
30°	...	491·1	1	491·1	5749·3	1	5749·3	·766	4404·0	·6428	3695·6
40°	...	361·2	1	361·2	4050·8	1	4050·8	·866	3508·0	·5000	2025·4
50°	...	291·3	1	291·3	3335·6	1	3335·6	·940	3135·4	·3420	1140·7
60°	...	263·6	1	263·6	3317·0	1	3317·0	·985	3267·2	·1736	575·8
70°	47·0	234·9	½	117·4	3586·5	½	1793·2	1·000	1793·2	·0000	000·0

Immersed wedge:

2903·3
−2301·2
2)602·1
301·0
× angular interval......... ·174
1·2040
21·070
30·10
52·3740
× longitudinal interval... 9
Excess in volume of immersion. .... = 471·3660

Both wedges — Statical stability:

3)26219·6
8739·8
× angular interval .... ·174
34·9592
611·786
873·98
1520·7252
× longitudinal interval 9
Moment of wedges uncorrected = 13685·1268
Correction for layer .... + 282·8400
÷ displacement .... 5425)13967·9668(2·57
E x = E F × sin. θ = ·78 × ·94 ...... ·78
·94
312
·702
·7332
E G .... = 2·57
− E x .... ·73
x G .... = 1·84
× displacement.......................... 155 tons.
9·20
92·0
184
Righting moment in foot tons = 285·20

Both wedges — Dynamical stability:

3)23497·6
7832·5
·174
31·3300
548·275
783·25
1362·855
9
12265·695
— 117·850
5425)12147·845(2·23
E F × vers. θ = ·78 × ·658
·78
5264
·4606
·51324
E_2 G = 2·23
— ·51
1·72 × 155 tons.
155
8·60
86·0
172
266·00*

* 266·00 = work done in foot tons.

Emersed Wedge.

0°	...	460·6	½	230·3
10°	...	399·0	1	399·0
20°	...	341·2	1	341·2
30°	...	300·0	1	300·0
40°	...	276·9	1	276·9
50°	...	275·9	1	275·9
60°	...	299·2	1	299·2
70°	56·3	357·5	½	178·7
				2301·2

Area of Inclined Water Section.

Products of Ordinates.
Immersed section .... 47·0
Emersed section .... 56·3
103·3
× longitudinal interval... 9
Area .... = 929·7 sq. ft.

Centre of Gravity of the Inclined Water Section.

Products of Squares.
Emersed .... 357·5
Immersed .... 234·9
2)122·6
61·3
× longitudinal interval 9
÷ area............... 929·7)551·70(0·593*

* 0·593ft. = the distance the centre of gravity of the inclined water plane is on the emersed side of the section.

Layer.

Excess in immersions ÷ area of inclined section =

929·7)471·36(·5

Thickness of layer = 0·5ft.

Correction for Statical Stability.

Volume of layer × distance C.G. is towards the emersed side = 471·4 × ·6 = 282·84 = correction.

Correction for Dynamical Stability.

471·4 volume of layer.
·25 = ¼ thickness of layer.
23·570
94·28
117·850 = correction.

Stability at 80° Inclination.

IMMERSED WEDGE.					BOTH WEDGES.						
								Statical Stability.		Dynamical Stability.	
Degrees of Inclination.	Products of Ordinates.	Products of Squares of Ordinates.	Multipliers.	Products.	Sums of Products of Cubes of Ordinates of both Wedges.	Multipliers.	Products for Moments of Wedges.	Co-sines of Angles of Inclination.	Products for Moments of Wedges × co-sine θ.	Sines of Angle of Inclination.	Products for Moments of Wedges × sine θ.
0°	...	460·6	1	460·6	7456·4	1	7456·4	·173	1289·9	·9848	7343·0
10°	...	525·3	4	2103·2	7471·7	4	29886·8	.342	10221·2	·9397	28084·6
20°	...	622·6	2	1245·2	7945·6	2	15891·2	·500	7945·6	·8660	13761·7
30°	...	491·1	4	1964·4	5749·3	4	22997·2	·642	14764·1	·7660	17615·8
40°	...	361·2	2	722·4	4050·8	2	8101·6	·766	6205·8	·6428	5207·7
50°	...	291·3	4	1165·2	3335·6	4	13342·4	·866	11554·5	·5000	6671·2
60°	...	263·6	2	527·2	3317·0	2	6634·0	·940	6235·9	·3420	2268·8
70°	...	234·9	4	939·6	3586·5	4	14346·0	·985	14130·8	·1736	2490·4
80°	43·6	204·8	1	204·8	4945·8	1	4945·8	1·000	4945·8	·0000	0000·0

Immersed wedge (continued):

9332·6
−8121·0

2)1211·6

605·8
× ½ angular interval...... ·058

4·8464
30·290

35·1364
× longitudinal interval... 9
Excess in volume of immersion. .... = 316·2276

Both wedges — Statical stability (continued):

3)77293·6

25764·5
× ½ angular interval.............. ·058

206·1160
1288·225

1494·3410
× longitudinal interval 9

Moment of wedges uncorrected = 13449·069
Correction for layer .... + 410·800

÷ displacement .... 5425)13859·869(2·55

E x = E F × sin. θ = ·78 × ·98 ...... ·78
·98

624
·702

·7644

E G ....= 2·55
− E x .... ·76

x G ....= 1·79
155 tons.

8·95
89·5
179

Righting moment in foot tons = 277·45

Both wedges — Dynamical stability (continued):

3)83443·2

27814·4
·058

222·5152
1390·720

1613·2372
9

14519·1348
— 47·4000

5425)14471·7348(2·66

E F × vers. θ
= ·78 × ·826
·78

6608
5782

E_2G = 2·66
·64 ·64428

2·02 × 155 tons.
2·02

3·10
310

313·10*

* 313·10 = work in foot tons.

EMERSED WEDGE.

0°	...	460·6	1	460·6
10°	...	399·0	4	1596·0
20°	...	341·2	2	682·4
30°	...	300·0	4	1200·0
40°	...	276·9	2	553·8
50°	...	275·9	4	1103·6
60°	...	292·2	2	584·4
70°	...	357·5	4	1430·0
80°	68·0	510·2	1	510·2
				8121·0

AREA OF INCLINED WATER SECTION.

Products of Ordinates.
Immersed section .... 43·6
Emersed section .... 68·0

111·6
× longitudinal interval... 9

Area .... = 1004·4 sq. ft.

CENTRE OF GRAVITY OF THE INCLINED WATER SECTION.

Products of Squares.
Emersed .... 510·2
Immersed .... 204·8

2)305·4

152·7
× longitudinal interval 9

÷ area..............1004·4)1374·3(1·368*

* 1·368ft. = the distance the centre of gravity of the inclined water plane is on the emersed side of the section.

LAYER.

Excess in immersions ÷ area of inclined section =

1004·4)316·227(·314

Thickness of layer = 0·314ft.

CORRECTION FOR STATICAL STABILITY.

Volume of layer × distance C.G. is towards the emersed side = 316· × 1·3 = 410·8 = correction.

CORRECTION FOR DYNAMICAL STABILITY.

316· volume of layer.
·15 = ½ thickness of layer.

15·80
31·6

47·40 = correction.

Stability at 90° Inclination.

Immersed Wedge. / Both Wedges.

Degrees of Inclination.	Products of Ordinates.	Products of Squares of Ordinates.	Multipliers.	Products.	Sums of Products of Cubes of Ordinates of both Wedges.	Multipliers.	Products for Moments of Wedges.	Statical Stability: Co-sines of Angles of Inclination.	Statical Stability: Products for Moments of Wedges × co-sine θ.	Dynamical Stability: Sines of Angle of Inclination.	Dynamical Stability: Products for Moments of Wedges × sine θ.
0°	...	460·6	1	460·6	7456·4	1	7456·4	·000	0·0	1·0000	7456·4
10°	...	525·8	3	1577·4	7471·7	3	22415·1	·173	3877·8	·9848	22074·4
20°	...	622·6	3	1867·8	7945·6	3	23836·8	·342	8152·2	·9397	22399·4
30°	...	491·1	2	982·2	5749·3	2	11498·6	·500	5749·3	·8660	9957·7
40°	...	361·2	3	1083·6	4050·8	3	12152·4	·642	7801·8	·7660	9308·7
50°	...	291·3	3	873·9	3335·6	3	10006·8	·766	7665·2	·6428	6432·3
60°	...	263·6	2	527·2	3317·0	2	6634·0	·866	5745·0	·5000	3317·0
70°	...	234·9	3	704·7	3586·5	3	10759·5	·940	10113·9	·3420	3679·7
80°	...	204·8	3	614·4	4945·8	3	14837·4	·985	14614·8	·1736	2575·7
90°	43·0	199·8	1	199·8	14754·9	1	14754·9	1·000	14754·9	·0000	0000·0
				8891·6					3)78474·9		3)87201·3

Both Wedges (continued).

	Statical Stability.	Dynamical Stability.
	26158·3	29067·1
× ⅓ angular interval	·0654	·0654
	10·46332	11·62684
	130·7915	145·3355
	1569·498	1744·026
	1710·75282	1900·98834
× longitudinal interval	9	9
Moment of wedges uncorrected	=15396·77538	17108·89506
Correction for layer	+ 484·40000	— 7·00000
÷ displacement	5425)15881·17538(2·9	5425)17101·89506(3·

The immersed wedge for this inclination being the smaller, it has to be subtracted from the emersed. (See below.)

$E\,x = E\,F \times \sin.\theta = \cdot78 \times 1\cdot = \cdot78$

E G .... =	2·90
− E x	·78
x G .... =	2·12
	155 tons.
	10·60
	106·0
	212
Righting moment in foot tons =	328·60

E F × vers. θ = ·78 × 1· = ·78

E_2G =	3·00
	·78
	2·22 × 155 tons.
	2·22
	3·10
	31·0
	310
	344·10*

* 344·10 = work in foot tons.

Emersed Wedge.

Degrees of Inclination.	Products of Ordinates.	Products of Squares of Ordinates.	Multipliers.	Products.
0°	...	460·6	1	460·6
10°	...	399·0	3	1197·0
20°	...	341·2	3	1023·6
30°	...	300·0	2	600·0
40°	...	276·9	3	830·7
50°	...	275·9	3	827·7
60°	...	292·2	2	584·4
70°	...	357·5	3	1072·5
80°	...	510·2	3	1530·6
90°	107·2	1241·3	1	1241·3
				9368·4
				− 8891·6
				2)476·8
				238·4
× ⅓ angular interval......				·0654
				9536
				1·1920
				14·304
				15·59136
× longitudinal interval...				9
Excess in volume of emersion .... =				140·32224

Area of Inclined Water Section.

	Products of Ordinates.
Immersed section	43·0
Emersed section	107·2
	150·2
× longitudinal interval...	9
Area	= 1351·8 sq. ft.

Centre of Gravity of the Inclined Water Section.

	Products of Squares.
Emersed	1208·0
Immersed	199·8
	2)1008·2
	504·1
× longitudinal interval	9
÷ area..........	1351·8)4536·9(3·356*

* 3·356ft. = the distance the centre of gravity of the inclined water plane is on the emersed side of the section.

Layer.

Excess in emersions ÷ area of inclined section =

1351·8)140·32224(·1

Thickness of layer = ·1ft.

Correction for Statical Stability.

Volume of layer × distance C.G. is towards the emersed side = 140 × 3·46 = 484·40 = correction.

Correction for Dynamical Stability.

140 volume of layer.
·05 = ½ thickness of layer.
7·00 = correction.

CALCULATING THE VOLUME AND CENTRE OF GRAVITY OF THE WEDGES OF IMMERSION AND EMERSION.

Volume of the Immersed Wedge.

From the table for 20°, page 393, take the sum of the products of squares of ordinates for the immersed wedge (3186·4) and divide it by 2. Then multiply by $\frac{1}{3}$ the angular interval ($= \frac{\cdot 174}{3} = \cdot 058$) and by the longitudinal interval = 9ft.

$$\frac{3186{\cdot}4}{2} \times \cdot 058 \times 9 = 831{\cdot}65 = \text{volume.}$$

That is, the volume of the wedge is 831·65 cubic feet. The volume of the emersed wedge would be similarly calculated.

Longitudinal Position of the Centre of the Wedge.

It must be premised that in the illustration which follows, the volume of the wedges has not been corrected, as explained in the tables. In practice for this calculation the water-line, as shown by the dotted line, Fig. 163, would be drawn, and the ordinates would radiate from the point *b*, instead of from *o*.

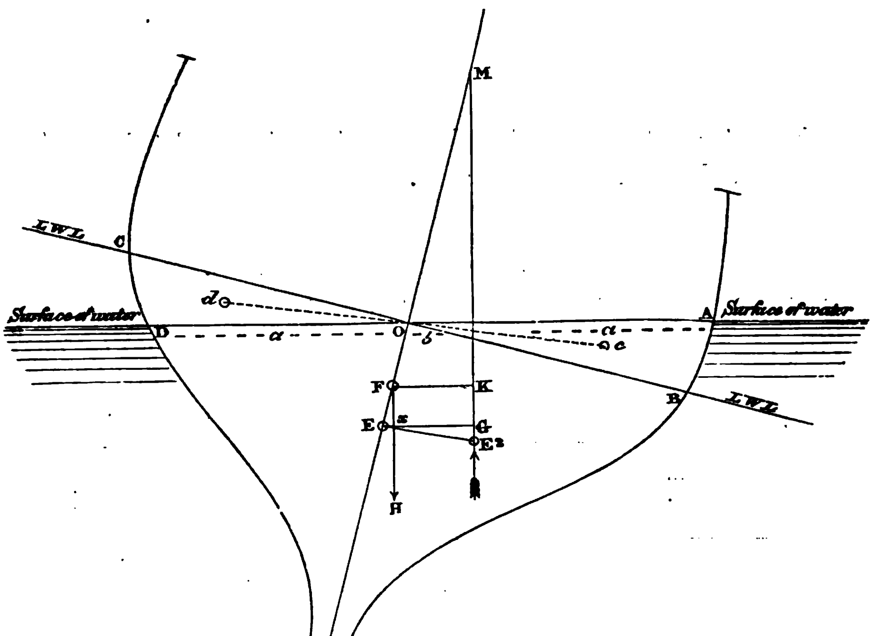


Fig. 163.

Take the "products of the squares of ordinates" from the tables for L.W.L. 10° and 20°, pages 392 and 393, and multiply them as shown in the example which here follows by the numbers, 1, 2, 3, 4, &c.

No. of Vertical Sections.	L.W.L. Section.			10° Section.			20° Section.		
	Products of Squares.	M.	Moments.	Products of Squares.	M.	Moments.	Products of Squares.	M.	Moments.
1	0·0	0	0·0	0·0	0	0·0	0·0	0	0·0
2	20·8	1	20·8	24·8	1	24·8	31·2	1	31·2
3	46·0	2	92·0	54·0	2	108·0	69·6	2	139·2
4	196·0	3	588·0	230·8	3	692·4	382·0	3	846·0
5	148·0	4	592·0	165·6	4	662·4	184·2	4	736·8
6	353·2	5	1766·0	368·4	5	1842·0	400·0	5	2000·0
7	172·2	6	1036·8	184·2	6	1105·2	196·0	6	1176·0
8	288·8	7	2021·6	324·0	7	2268·0	360·8	7	2525·6
9	92·4	8	739·2	121·6	8	972·8	148·0	8	1184·0
10	64·0	9	576·0	104·0	9	936·0	196·0	9	1764·0
11	00·0	10	00·0	0·0	10	0·0	0·0	10	0·0
	2)1382·0		2)7432·4	2)1577·4		2)8611·6	2)1867·8		2)10402·8
	691·0		3716·2	788·7		4305·8	933·9		5201·4

L.W.L....	691·0	3716·2
10°	788·7	4305·8
20°	933·9	5201·4

2413·6)13223·4(5·47 × Longitudinal interval.
12068·0   9

1155·40   49·23 = the distance the centre of the wedge is from the stem.
965·44

189·960
168·952

21·008

The centre of the emersed wedge was similarly calculated, and found to be 48 feet abaft the foreside of stem.

Transverse Position of the Centre.

Take the sums of the cubes of the ordinates from the tables for L.W.L. 10° and 20°, pages 392 and 393, and operate upon them as follows:

	Sums of cubes of ordinates	Cosines.	Products.		Products.
L.W. L....	3728·8	1·00	3728·8	1	3728·8
10°	4454·2	·98	4365·1	4	17460·4
20°	5511·3	·94	5180·6	1	5180·6

26369·8
× $\frac{1}{3}$ angular interval = $\frac{\cdot 174}{3}$ = ·058

210·9584
1318·490

1529·4484
× $\frac{1}{3}$ longitudinal interval = $\frac{9}{3}$ = 3

+ volume wedge 831·6)4588·3452(5·5ft.
4158·0

430·34
415·80

That is, the centre of the wedge is 5·5ft. from the point *b*, Fig. 163.

The centre of the emersed wedge would be similarly calculated, using the volume of that wedge as the divisor.

VERTICAL POSITION ON THE CENTRE OF THE WEDGE.

This calculation only varies from the preceding by using the sine of each angle instead of the co-sine.

	Sums of cubes of ordinates.		Products.		Products.
L.W.L. ...	3738·8	·00	0000	1	0000
10°	4454·2	·17	757	4	3028
20°	5511·3	·34	1874	1	1874

$$\begin{array}{rr}
 & 4902 \\
\times \tfrac{1}{3}\text{ angular interval} = & \underline{\cdot 058} \\
 & 39{\cdot}216 \\
 & \underline{245{\cdot}10} \\
 & 284{\cdot}316 \\
\times \tfrac{1}{3}\text{ longitudinal interval} = & \underline{\quad 3} \\
\div\text{ volume wedge } 831{\cdot}6) & 852{\cdot}948(1{\cdot}02 \\
 & \underline{831{\cdot}6} \\
 & 21{\cdot}348
\end{array}$$

That is, the centre of the wedge is 1·02ft. from the L.W.L.

If the centre of the wedges be thus determined, the stability can be ascertained according to the principles explained on pages 23, 24, &c.

CALCULATION OF THE EFFECT OF SHIFTING AND REMOVING WEIGHTS ON STABILITY.

It frequently becomes necessary to remove a portion of a vessel's weights, and place it in a different position as to depth. For instance, a portion of the loose ballast which Sea Belle has might be taken from its present situation and added to her lead keel. It is obvious that, if this were done, the centre of gravity of the vessel would be shifted to some lower position. By the rule explained on page 331, this position can readily be determined. Thus: It will be presumed that the 6·3 tons loose ballast of the Sea Belle has been taken out and placed 6ft. lower down, underneath or about the keel; then the distance the centre of gravity has been moved will be $x = \frac{W \times d}{s}$, where W the weight shifted, d the distance it has been shifted, s the whole weight of the vessel, and x the distance the centre of gravity has been shifted; or by stating the exact values, $x = \frac{6{\cdot}3 \text{ tons} \times 6\text{ft.}}{155 \text{ tons}} = {\cdot}24$ft. Thus, the centre of gravity of Sea Belle would be shifted ·24ft., or nearly 3in. lower, by the removal of 6·3 tons to her keel.

But the object of shifting the weights may not always be simply to obtain a lower situation of the centre of gravity; it may be to dispense with some of the other weights altogether, to reduce the displacement, and at the same time insure that the original stability or righting moment is secured.

It has been shown that by shifting 6·3 tons of ballast to a greater depth the centre of gravity was lowered ·24ft., bringing the meta-centric height (or height of meta-centre above the centre of gravity) from 3·3ft. to 3·54ft.; the righting moment in foot tons at 20° inclination would, therefore, be with the latter meta-centric height $3{\cdot}54 \times 155 \times {\cdot}342 = 187{\cdot}6$ foot tons. The righting moment with the centre of gravity in its original position was 174 foot tons, and the object now will be to ascertain to what extent weight can be taken from the inside of the hull, in the shape of ballast, so as to retain the original moment of 174 foot tons.

As the position of the centre of buoyancy and meta-centre for different displacements can only be ascertained by separate calculations, the process of calculating the exact quantity to be removed would be rather tedious, and a rough way of approximating the quantity will be resorted to. To begin, it will be best to ascertain what weight can be further removed to restore the centre of gravity to its original position, and then take half that weight as the quantity to be removed.*

Let it be assumed that a portion of ballast, such as the limber ballast, whose centre of gravity would be 3·7ft. below the centre of gravity of the vessel (in its new position) could be removed: then the exact weight to be thus removed will bear the following proportion to the 6·3 tons already shifted a distance of 6ft. to the keel:—That is, as 3·7ft. is to 6ft., so is 6·3 tons to the quantity to be removed, which will be termed x; then $3{\cdot}7 : 6 :: 6{\cdot}3 : x = \frac{6 \times 6{\cdot}3}{3{\cdot}7} = 10$ tons. Half of this quantity will be taken for removal. If 5 tons be so taken away, its effect on the new centre of gravity of the vessel will be $\frac{3{\cdot}7 \times 5}{155} = {\cdot}12$; or, ·12ft. is the distance the centre of gravity would be brought back towards its original position, making the meta-centric height 3·42ft.

The displacement would thus be 150 tons, and the righting moment at 20° inclination will be $150 \times 3{\cdot}42 \times {\cdot}340 = 174$ foot tons. Thus 5 tons could be removed from Sea Belle's displacement if 6·3 tons were also taken from the inside and placed on her keel, and she would retain an equal righting power.

Of course this calculation is not strictly accurate, as a new centre of buoyancy and meta-centre ought to be calculated for the altered immersion and displacement. This calculation would involve some trouble, and in the end would not be very much more accurate when the weight removed

is not more than ·04 of the total weight. The extent she will rise bodily out of the water can be read off from a scale of displacement (see Fig. 158) or it can be approximated by dividing the quantity removed by the displacement per inch of immersion at the load water-line. For Sea Belle the displacement per inch at the L.W.L. is 2·64 tons; then $\frac{5}{2 \cdot 64} = 1 \cdot 8$in. $= \cdot 158$ft.

If an extra quantity of ballast is to be put on board then its distance below the centre of gravity must be ascertained, and its effect calculated thus. Say the weight to be put on board is 5 tons, and its distance below the centre of gravity 3·7ft., and the increased displacement 160 tons, then $\frac{5 \times 3 \cdot 7}{160} = \cdot 116$ = the distance the C.G. has been lowered or from 3·3 to 3·416ft. The new righting moment at 20° will therefore be $160 \times 3 \cdot 416 \times \cdot 342 = 187$ foot tons.

AREA AND CENTRE OF EFFORT OF SAILS.

The calculations for finding the areas of sails and determining their "centre of effort" are very simple. In fact, the centre of effort is found by an almost mechanical operation; and, although this may appear to be a rather complex affair upon merely looking at the sail plan (Plate XI.), it will not be found to be so if its intricacies be unravelled with patience.

The plate also shows how the measurements to calculate the areas of the various sails are taken and will be easily understood.*

If a straight edge be applied to the after leach of the mainsail it will be found to have a considerable convexity, and to allow for this a line is produced which cuts the curved edge of the sail as shown by the ticked line A B (Plate XI). This ticked line will be the one used in the calculation, and the curved line will be wholly disregarded.

The area of the sails must be first calculated, beginning with the mainsail. Divide the mainsail into two triangles—A B C and A C D. The area of each of these triangles will be found by the rule given on page 326. Let a be set off at right angles to A B from C, then $\frac{A\,B \times a}{2}$ = area of the triangle A B C. Thus, for A B we have 71ft.; for a 43ft.; then

```
     71
     43
    ---
    213
   284
   ----
2)3053
  ------
  1526·5 sq. ft. = area.
```

* It should be noted that the head of the mainsail is shown much flatter than is usual. The gaff usually makes an angle of 53° with the horizon.

Then for the triangle A C D we have for A C 74·5ft.; for b 35·5ft.; then

$$\begin{array}{r l}
74{\cdot}5 & \\
\underline{35{\cdot}5} & \\
37{\cdot}25 & \\
372{\cdot}5 & \\
\underline{2235\phantom{{\cdot}00}} & \\
2)\underline{2644{\cdot}75} & \\
1322{\cdot}37 & \text{sq. ft.} = \text{area.} \\
\underline{1526{\cdot}50} & \\
\underline{2848{\cdot}87} & = \text{whole area of mainsail.}
\end{array}$$

Next the area of the foresail will be obtained; but as it is of triangular form no division of the sail is necessary. The luff of the sail will serve as the base of the triangle, and will be represented by E F, and the foot by F G; then c will be set off at right angles to E F to meet G. The length of E F is 50ft.; and c 27ft.; then

$$\begin{array}{r l}
27 & \\
\underline{50} & \\
2)\underline{1350} & \\
\underline{675} & \text{sq. ft.} = \text{area of foresail.}
\end{array}$$

The jib being triangular in form will have its area found in a precisely similar manner. The base of the triangle will be the line H I: from K produce the line d at right angles to H I, as in the other examples. The length of the line H I is 7·5ft., and the length of d is 23·5ft.; then

$$\begin{array}{r l}
23{\cdot}5 & \\
\underline{7{\cdot}5} & \\
11{\cdot}75 & \\
\underline{164{\cdot}5} & \\
2)\underline{176{\cdot}25} & \\
\underline{88{\cdot}12} & \text{sq. ft.} = \text{area of jib.}
\end{array}$$

The round in the foot of the jib was disregarded; that is, it was assumed that a straight line was produced from I to K.

To find the area of the topsail it must be partitioned as was the mainsail; thus we have L M N forming one triangle, and L N O the other. The side L N = 37·3ft.; the perpendicular e = 39ft.; then

$$\begin{array}{r l}
37{\cdot}3 & \\
\underline{39} & \\
335{\cdot}7 & \\
\underline{1119\phantom{{\cdot}0}} & \\
2)\underline{1454{\cdot}7} & \\
\underline{727{\cdot}35} & \text{sq. ft.} = \text{area.}
\end{array}$$

For the triangle L N O we have L O = 43·5ft.; and f = 22·5ft.; then

$$\begin{array}{r}
43{\cdot}5 \\
\underline{22{\cdot}5} \\
21{\cdot}75 \\
87{\cdot}0 \\
\underline{870\phantom{{\cdot}0}} \\
2)978{\cdot}75 \\
489{\cdot}37 = \text{area.} \\
\underline{727{\cdot}35} \\
\underline{1216{\cdot}72} \text{ sq. ft. = whole area of topsail.}
\end{array}$$

Centre of Effort of Sails.

The rules for finding the centre of gravity of such figures as the mainsail, jib, and foresail have already been described on page 332, and we can proceed at once to show the application of these rules.

Mainsail.

The mainsail A B C D (Plate XI.) is divided into four triangles by the line drawn from A to C, and by the line drawn from B to D. The centre of gravity of each of these triangles will be found. Taking that of the triangle A B C first: the side A B is bisected in *a* (*i.e.*, divided in the middle), from the point of bisection produce a line to C. [In practice this line need not be drawn right through, but only just over that part of the figure in which the judgment assumes the centre of gravity to lie.] Next bisect A C in *b*, and produce a line from the point of bisection to B. The point *x* 1, where the two lines from *a* and *b* intersect, will be the centre of gravity of the triangle.

The centre of gravity of other triangles will be found by exactly the same process. Take the triangles A C D and bisect the lines A C in *b* and A D in *c*. [It will be found that A C has already been bisected for the triangle A B C, whose centre of gravity was previously found.] Produce lines from *b* to D and from *c* to C, and the point *x* 2 where they intersect will be the centre of gravity of the triangle A C D.

Next we have the two triangles A B D and B C D to deal with. Taking B C D first, bisect the line D C in *d*, and from the point of bisection produce a line to B; then bisect B D in *e* and produce a line to C. The intersection at *x* 3 will be the centre of gravity of the triangle B C D.

For the triangle A B D the necessary bisections have been made at *a*, *e*, or *c*, and all that remains to be done is to produce lines from *a* to D, and from *e* to A. [To test the accuracy of the result a further line could be

produced from *c* to B.] The point of intersection, *x* 4, will be the centre of gravity of the triangle A B D. The centre of gravity of the whole figure (*i.e.*, the mainsail) will be found by drawing a line from *x* 1 to *x* 2; then join *x* 3 and *x* 4 in a similar way, and the point *x* 5 where the two lines intersect will be the centre of gravity of the figure A B C D, that is to say, the centre of gravity of the mainsail.

Foresail.

The centre of gravity of the foresail will be found by bisecting E F in *f* and E G in *g*, and producing a line from *f* to G and from *g* to F. The point of intersection, *x* 6, will be the centre of gravity of the foresail.

Jib.

The centre of gravity of the jib will be ascertained from bisecting H I in *h* and H K in *i*; then having produced a line from *h* to K and from *i* to I, the point *x* 7 where the lines intersect will be the centre of gravity of the jib.

Topsail.

To find the centre of gravity of the topsail that sail must be divided into four triangles, as was the mainsail. First taking the triangle L M N, bisect L N in *j* and produce a line from *j* to M; then bisect M N in *k* and produce a line from *k* to L; the point of intersection; *x* 8, will be the centre of gravity of the triangle L N O.

For the triangle L N O bisect L N in *j* and L O in *l*, produce a line from *j* to O and from *l* to N, and their point of intersection, *x* 9, will be the centre of gravity of the triangle L N O.

The centre of gravity of the two other triangles M N O and L M O must next be found to complete the operation. Proceeding with the triangle M N O, bisect M O in *m* and produce a line to N; then bisect M N in *k* and produce a line from *k* to O; the point of intersection, *x* 10 will be the centre of gravity of the triangle M N O.

Then proceeding with L M O, bisect M O in *m* and L O in *l*; produce a line from *m* to L and from *l* to M; the point of intersection, *x* 11, of these lines will be the centre of gravity of the triangle L M O.

To find the common centre of gravity of the whole figure (*i.e.*, the topsail) join *x* 8 and *x* 9; then join *x* 10 and *x* 11; then the point of intersection at *x* 12 will be the centre of gravity of the topsail.

Having found the various centres of the sails, calculation must be resorted to for ascertaining the common centre of gravity of the whole sail plan, generally termed "centre of effort" of the sails; that is, if the

whole force of the wind exerted on the sails were concentrated in one point, that point would be at the common centre of gravity of the sails.

In the first place a perpendicular, P, must be erected on the L.W.L. at the point where the latter is cut by the stem. The distance the centre of gravity of each of the sails is from the perpendicular P will then be measured in a horizontal direction, that is, at right angles to P, as shown by the dotted lines P *x*. Thus P *x* for the mainsail is 61·2ft.; for the topsail, P *x* 1, is 44·5ft.; for the foresail, P *x* 2, is 20·7ft. All these distances are *abaft* the perpendicular P; the distance P *x* for the jib is very small, only 0·3ft., and is *forward* of the perpendicular.

These distances have now to be multiplied into the area of the sails, and the products or moments will be summed as hereafter shown, and divided by the whole area of the sails.

Mainsail	area	×	P *x*	=	2849 sq. ft.	×	61·2ft.	=	174359
Topsail	area	×	P *x* 1	=	1217 sq. ft.	×	44·5ft.	=	54156
Foresail	area	×	P *x* 2	=	675 sq. ft.	×	20·7ft.	=	13973
Jib	area	×	P *x* 3	=	881 sq. ft.	×	0·3ft.	=	264

The moments which are abaft P (that is, those of the mainsail, foresail, and topsail) will now be added together; from the sum thus obtained will be *subtracted* the moment for the jib, the centre of gravity of the sail being *forward* of P. The corrected moment thus found will be divided by the *whole* area of the sails, and the quotient will give the distance the centre of gravity (*i.e.*, centre of effort) of the sails is from P.

		Areas.				Moments.
Mainsail	=	2849 sq. ft.	×	61·2	=	174359
Topsail	=	1217 sq. ft.	×	44·5	=	54156
Foresail	=	675 sq. ft.	×	20·7	=	13973
						242488
Jib		881 sq. ft.	×	0·3	=	264–
Total area		5622				5622)242224(43·
						22488
						17344
						16866
						478

That is, the centre off effort of the sails is 43ft. from the perpendicular P, represented on the sail plan (Plate XI.) by the circle s C E.

It is usual to calculate the centre of effort without the topsail; in this instance the result without the topsail is:

		Areas.				Moments.
Mainsail	=	2849 sq. ft.	×	61·2	=	174359
Foresail	=	675 sq. ft.	×	44·5	=	13973
						188332
Jib	=	881 sq. ft.	×	0·3	=	264–
Total area		4405				4405)188068(42·69

That is, the centre of effort of the lower sails is 42·7ft. abaft P, or abaft the fore side of the stem at the L.W.L., represented on the sail plan by the circles C E 2.

The centre of effort of the sails as to height above the load water-line has next to be found. The distance the centre of gravity of each sail is above, and at right angles to, the L.W.L. will be measured. These distances, represented by P *x* 4, &c., will be multiplied by the respective areas of the sails; the *whole* of the moments thus found will then be summed and divided by the whole area of sails. For example

		Areas.				Moments.
Mainsail	=	2849 sq. ft.	×	34·7	=	98860
Topsail	=	1217 sq. ft.	×	78·7	=	95778
Foresail	=	675 sq. ft.	×	21·2	=	14310
Jib	=	881 sq. ft.	×	25·2	=	22201
		5622				5622)231149(41·1
						22488
						6269
						5622
						647

That is, the centre of effort of the sails, including topsail, is 41·1ft. above the load water-line shown by the circles C E.

The centre of effort above the load water-line has now to be calculated without the topsail.

	Moments.
Mainsail	98860
Foresail	14310
Jib	22201
Area of lower sails =	4405)135371(30·73

That is, the centre of effort of the lower sails is 30·74ft. above the load water-line shown by the circle C E 2.

The centre of effort of a yawl's sails would be found in a similar manner, excepting that the mizen would have to be considered. As the mizen will be in form like the mainsail, no explanation is necessary as to finding its area and centre of gravity. When the centre of gravity of the mizen is found, its distance from P will be measured, and this distance will be multiplied into the area and form part of the moments of the calculation for determining the centre of effort of the sails.

The centre of effort of a schooner's sails will be calculated in the same way as a yawl's, the mainsail for the purpose of the calculation being in the position of the mizen.

It should be remembered that in *all cases* if there are moments *forward* of P they must be subtracted from those which are *abaft* P before the sum is divided by the *whole* area of sails.

Sometimes the perpendicular P is erected exactly over the centre of lateral resistance of the vessel; in such a case a large proportion of the moments would be *forward* of P, and, consequently, would require to be subtracted.

It is of little consequence where P is erected so far as the result which determines the position of the centre of effort goes; but less error is likely to occur if the perpendicular is so placed that a minimum of moments has to be deducted; and the perpendicular P might be erected say at the extreme fore end of head sail.

If it be necessary to know the height the centre of effort is above the centre of lateral resistance or above the centre of buoyancy, the distance the centre of lateral resistance or centre of buoyancy is below

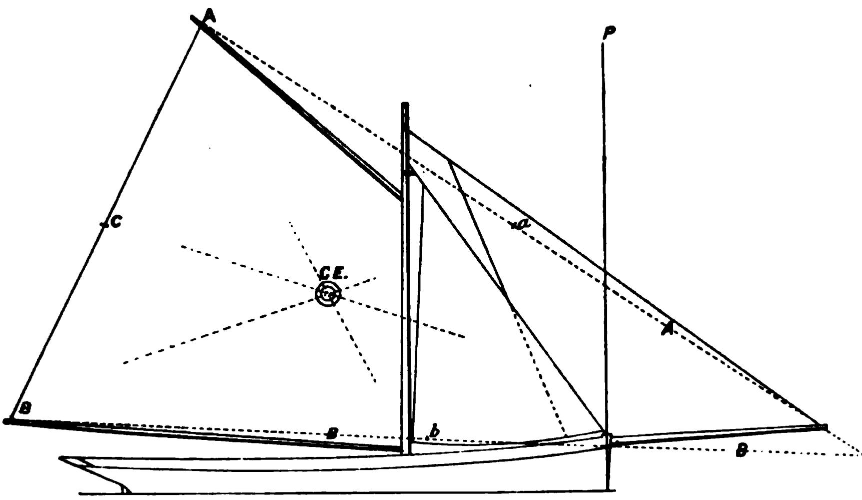


FIG. 164.

the L.W.L. will be added to the distance the centre of effort is above the L.W.L.

A rough method of approximating the position of the centre of gravity of a cutter's sails is to assume that the head sails and main sail form one triangle, as shown by the dotted line (Fig. 164); but unless the boundaries are made with care the results are likely to be very incorrect.

It will be seen that the boundary line comes below the bowsprit to compensate for that part above the luff of the jib or gaff.

The area of lower sail could be roughly calculated from such a diagram, but the method does not commend itself if accuracy is desired.

The line B is bisected in *b*, and A in *a*; also A B in *c*. Lines are then drawn from *a* to the angle at B, and from *b* to the angle at A, and

from *c* to the angle below the bowsprit end. Where the lines intersect at C E will be the centre of effort approximately.

On page 100 the practice of placing the centre of effort ahead of the centre of lateral resistance is explained as ranging from ·01 to ·03 of the length on load line. The latter ratio will not be found too great for vessels which have their greatest draught about amidships, and they carry the depth well forward before the forefoot much rounds up.

That is to say, if the length on the load line be 41ft., then 41 × ·03 = 1·23ft. In some cases this distance might be even exceeded, especially in vessels whose mainsails bear a very large proportion to the total area of lower sail.

CHAPTER XV.

DESIGNING.

Instruments used in Drawing, and their Management.

If the draughtsman has a knowledge of geometrical drawing he will find it of much service in delineating upon paper the various problems connected with ship designing. However, with the exercise of patience and perseverance, anyone unacquainted with such drawing will be able to draw the various parts and sectional outlines of a vessel with facility. In the first place the necessary instruments must be obtained,* and it is important that all these instruments should be accurately made. The following enumeration will comprise the necessary "outfit" for a draughtsman.

A large drawing board, 6ft. or 8ft. in length by 3ft. in breadth; the sides of this board should be true, as it may be occasionally necessary to "square" from it.

One straight edge 6ft. in length; one 3ft. in length; and one 1ft. 6in. in length.

One ⅂ square 2ft. × 1ft. 3in. Two or three 45° squares (these can be obtained in vulcanite). By a 45° square is meant a right angled triangle; where one of the angles is a right angle and the other two angles of 45°.

A protractor for setting off angles. It will be found very convenient for setting off angles, and for squaring, to have a square with two equal sides, 1ft. each, made in the form of a quadrant. The arc will be divided into degrees and minutes. Such an instrument can be used for drawing perpendiculars or squaring, or for the setting off of angles.

One parallel rule, 1ft. in length. If two are had, one should be 8in. long, and the other 1ft. 6in.

Several scales of feet. The usual scales are $\frac{1}{8}$in. = foot; $\frac{1}{4}$in. = foot; $\frac{3}{8}$in. = foot; $\frac{1}{2}$in. = foot; $\frac{5}{8}$in. = foot; $\frac{3}{4}$in. = foot; $\frac{7}{8}$in. = foot; 1in. = foot.

* All the instruments required by naval architects can be obtained at Stanley's Depôt, Great Turnstile, Holborn, London. The Admiralty and Royal School of Naval Architecture are supplied from this establishment.

Each scale is divided into twelfths to represent divisions of the foot by inches; the underside of the scale should be divided into tenths to represent the decimal parts of a foot. Of course any odd kind of scale can be made to suit particular cases. A scale 3ft. long containing *all* the scales (marked in twelfths) should also be had for facility and accuracy in measuring great lengths.* In taking measurements for constructing the drawing all fractional parts of a foot are described as inches; but in taking measurements for calculations the odd inches are regarded as decimal fractions of a foot.

A small box of drawing instruments containing compasses or dividers (with movable legs) and two or three bow pens, must be obtained, or the instruments can be bought separately. A bow pen with an extra blade, termed a section pen, will be found very convenient for various special purposes. The "extra" blade is a little shorter than the other two, and serves as a guard to keep the pen a small distance from the batten or straight edge (when the bevelled edge of the latter, as it always should be, is uppermost). It can be regulated as to its distance from the other two blades by a small screw—thus it will be found very useful in drawing the double lines representing the deck sheer and rail of a vessel.

For reducing or enlarging a drawing a pair of "proportional compasses" will be required. They are made with a sliding joint so that they can be regulated as required; that is to say, if two of the legs are opened 2in. the other two will be opened 4in.; or if one set of legs be opened 4in. the other will be opened 3in., and so on, one set of legs being made to bear any required proportion to the others.

For the drawing of curved lines a set of flexible battens will be required. Those made of lancewood will be found to be the best. The battens will vary in length, stiffness, and size. For the sheer line a batten 5ft. or 6ft. in length by $\frac{1}{8}$in. by $\frac{3}{16}$ in. at one end and gradually tapering towards the other end. A batten of the same length, but half as square, will be required for the water-lines. Another long batten tapering at each end will be found useful for sweeping in bow and buttock lines. A corresponding set of battens about 3ft. in length should also be obtained. For drawing lines which have abrupt curves very fine spline battens must be used, and these may vary in length from nine inches to two or three feet. These splines should be a bare $\frac{1}{8}$in. wide by $\frac{1}{32}$in. thick.

The battens are fixed by means of lead weights, and from eighteen to twenty-four of these should be obtained. Twelve should weigh 6lb. each, and the others may vary from 3lb. to 4lb. each. The most con-

* Cardboard scales, with the feet divided into twelfths, can be obtained at Holtzapffel and Co.'s, Charing-cross. They are 1s. each, and being "machine-divided" are very accurate.

venient form for the weights is that depicted by Fig. 165. Fig. 166 shows the under side of the lead weight.

These weights are sometimes cased in mahogany or mounted on brass, but a cheaper and equally good plan is to fix them on a mahogany plate, which terminates in a point, as shown. It is this point which rests on the batten to keep it in shape. A good proportion for the large weights will be 2¼in. high, 1¾in. broad at the shoulder, and 1⅛in. at the thick end. The extreme length of the lead should be 5¼in. The lighter weights should be very much narrower, and should not exceed 1⅛in at the broadest part at the shoulder, but they need not be less than 2in. or 2¼in. in height, and 5in. in length. It is necessary that the wood upon which the weights are fixed should be hard; otherwise, the point which has to rest upon the batten will soon wear away, and there will then be difficulty in holding the batten. The wood on the bottom of the lighter weights (which will be used for the splines) should project 1¼in. beyond the lead at the point, gradually tapering (see Fig. 165), and project $\frac{1}{16}$in. at the sides and end.

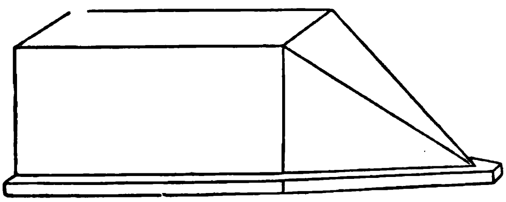


Fig. 165.

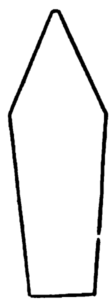


Fig. 166.

It is usual to have a number of curves of different shapes, known as shipwrights' curves and French curves; but the most useful form are those which are pear-shaped, and made by Mr. Stanley under the name of "Dixon Kemp." With a set of pear-shaped curves or moulds, any conceivable curve can be drawn; and the particular mould to suit a special curve can be more readily selected than could the suitable French curve. There are usually eight in a set of pear-shaped curves, varying in length from 1ft. to 3¼in. The edges of these curves should be bevelled.

It was at one time usual to use curves or moulds for the vertical sections shown in the body plan; but the small spline battens are now more generally used. In the first place, if moulds are used, the curves must be drawn in several pieces, and unless the draughtsman be expert in the use of the pen or pencil, the various joints will show in the drawing; if a spline be used, the whole curve can be swept in at once, and the line

will be unbroken and cleaner than it will be when done by the more tedious process with moulds. In bending the thin battens round the curves of the sections of the body plan, or round any abrupt curves, they must be handled very gently and patiently, otherwise the batten will be broken, or it will fly out of position. When the curves are so abrupt, as they will sometimes be in small drawings, that the spline cannot be bent round them, the moulds will be resorted to. Before the beginner attempts to define a curved line by putting a batten over spots alone, he should copy one or two body plans by means of tracing paper; he will thus become acquainted with the peculiarities of battens, and discover to what extent they must, in producing "fair curves," be allowed to take their natural curves and inflexions, or to what extent they are sometimes forced out of these natural curves and inflexions.

One part of a curve may be "fair" with another part of the curve, although the batten may require some humouring to get it round the lines. This will frequently be found to be the case when lines have contrary flexures.

When the batten has been secured in its proper position round the curve by the lead weights, the draughtsman must pass his eye along the batten to see that none of the weights overlap or protrude beyond the batten; if they do so overlap, the pen might be thrown out, and a very ugly scrawl would then occur on the drawing. If the section pen before referred to be not used, great care must be taken to avoid the point getting underneath the batten; if it does so get underneath, all the ink will be instantly extracted from it, and the line will consequently be blurred.* No parts of a drawing should be inked in until it is completed as a whole in pencil.

The Construction Drawing.

In representing a vessel on paper, it is usual to have three projections: the Sheer Plan, the Half-breadth Plan, and the Body Plan (see Plate XII.)

The Sheer Plan represents the longitudinal and vertical section of the vessel. On it the water-lines appear as straight lines, and the buttock lines and bow lines as curves. If the drawing be to the moulding of the frames (that is, without the plank), the water-lines will terminate in the rabbet of the stem aud stern-post; if the drawing be to the outside of

* Many useful hints on the management and use of drawing instruments, &c., will be found in a little work called "The Workman's Manual of Engineering Drawing," published by Lockwood.

The best paper is that known as "continuous drawing paper." It can be obtained 5ft. or 6ft. wide, in 100yd. lengths, of Messrs. Waterlow, 24, Birchin-lane, London, E.C.

the plank, the water-lines will be continued to the extreme edge of the stem and stern-post.

The Half-breadth Plan represents the longitudinal and horizontal section of the vessel; on it the water-lines are curves, and the buttock and bow lines straight lines.

The Body Plan represents transverse vertical sections of the vessel, and on it the water-lines, diagonal lines, buttock, and bow lines appear as straight lines. (The water-lines, however, may not necessarily be straight lines; if the keel of the vessel be made the base line, and if the keel be not parallel to the load water and other water-lines, then the water-lines will show with some curvature in the Body Plan. However, so far as yacht building is concerned, the general practice is to make the load-water line the base line.)

From the sheer plan are obtained the heights (the freeboard) for the vertical sections above the water, and the depths below. The ending of the water-lines at the stem and at the stern-post; and the true shape of the buttock lines and the bow lines.

From the half-breadth plan are obtained the half breadths for the deck and the half breadths for the water-lines; and the ending of the bow line by its intersection with the deck line; and the ending of the buttock line at the stern.

Knowing the heights, depths, and breadths of the vertical sections and the depths of the buttock lines, the body plan can be constructed.

It is frequently the practice to complete the sheer drawing first, but there is no reason for so doing, and it may be often found more convenient to draw each plan by stages, as one part interprets, or is required to interpret, another. It was found convenient to adopt the method in the description which follows, but it is not intended to be arbitrary. Indeed, the description given is only intended to assist those who have no knowledge of designing, and the young designer as he gains experience will vary the procedure according to his judgment or convenience.

It will be assumed that the displacement of the vessel is to be about 14 tons, and length on load line 36ft., and breadth 8ft. The drawing in this case will be made with the plank on; and, as a rule, this is the more convenient plan, although subsequent trouble will be incurred in taking off the plank for laying down in the mould loft.

Take the straight edge and produce the line A B at some convenient distance from the lower edge of the paper; this line will be the base line of the half-breadth plan, and all ordinates will be drawn at right angles to it. Having determined that the length of the vessel shall be 36ft. on the water-line from the fore side of the stem to the aft side of the stern-

post, and the extreme beam 8ft., then set off the *length* on the line at A B from *a* to *b*.

Next take the square, and from the points *a b* erect two perpendiculars C and D. On these perpendiculars set off F and G at convenient equal distances from *a* and *b*, as the positions for the load water-line on the sheer plan, which will thus be drawn parallel to the middle line of the half breadth plan. The distances between *a b* and F G will equal the half breadth and the draught of water of the vessel; but the distance had better be a little greater, or the keel might come inconveniently near the water-lines or deck line of the half breadth plan. Say the half beam of the vessel is 4ft. and the draft of water 7ft.; this will make 11ft., so 13ft. can be taken as a convenient distance between *a b* and F G. With the straight edge draw the line through L.W.L., and produce it far enough to left to form the load water-line of the body plan as well.

Next the position of the midship section must be determined; first find the centre of length of the load line at *g*, and having determined that the midship section shall be, say, ·05ft. of the length on the load line abaft the centre of length, multiply 36 by ·05 = 1·8ft.

[If the mid-section is to have considerable rake (see page 177), the greatest half breadth on the load water-line might come farther aft than the position assigned by the factor ·05.]

Set off this distance from *g* to ♓, and drop a perpendicular from ♓ to the half breadth plan also marked ♓. This will be the position of the midship section in the vessel.

Measure the distance ♓ *b*, and divide it into eight equal intervals by the ordinates 1, 2, 3, 4, 5, 6, 7. In this case the length is 19·8ft. which, divided by eight, gives intervals of 2·475ft. each, or as nearly as possible 2ft. 6in. The ordinates at these intervals should be drawn as lightly as possible, as they are only temporary contrivances for assistance in drawing the load water line, and may be subsequently removed.

[If the drawing is to be made without the plank, then the half siding of the stem, rabbet, and boarding must be drawn on the half breadth plan. (See "Laying Off.") Say the half siding (*i.e.*, half thickness) of the stem is to be 2in., and the moulding of it (*i.e.*, its width from its fore side to aft side) 7in.; set off these distances at *b*, and the rabbet will be found when the water lines are drawn by knowing the thickness of the plank, and setting that thickness off at right angles to the water-line. It should be noted that the appearance of the lines without the plank will be somewhat different to what they would be with the plank on; this will be especially noticeable in the load water-line aft.]

The beam on the load line, to allow for a little roundness of side, has been fixed at 7ft. 7in. The half beam on the load line will therefore be 3ft. 6in. Set out this distance on the line ⊗ M for the greatest half breadth, the load water line in the half breadth plan.

It will be assumed that the factors on page 173 have been selected for computing the ordinates of the fore part of the load water-line, and the greatest half breadth will be consecutively multiplied by them.

	Greatest half breadths.		Multipliers.		Length of ordinates.
7.	3·6ft.	×	·9833	=	3·54ft.
6.	3·6ft.	×	·9183	=	3·31ft.
5.	3·6ft.	×	·8125	=	2·92ft.
4.	3·6ft.	×	·6700	=	2·41ft.
3.	3·6ft.	×	·5000	=	1·80ft.
2.	3·6ft.	×	·3200	=	1·15ft.
1.	3·6ft.	×	·1500	=	0·54ft.

These distances will be set off on the half breadth plan on the ordinates 1, 2, 3, 4, 5, 6, 7; a small spot, or cross, or other mark, being made with the pencil at each point. The batten must now be taken and bent round the spots—not *over* them, but just clear inside, so as to admit the pencil going over them. If the ordinates have been carefully calcu-

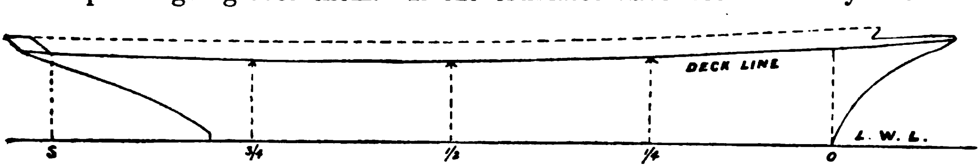


Fig. 167.

lated and set off the batten will take all the spots in, and the curve will be a fair one.

It will be convenient at this stage to draw the sheer of the vessel which represents the deck line. There is no rule as to what sheer should be given, but generally it is considered that the vessel is sufficiently high at the bow if the height there exceeds the height at ⊗ as 1 to 1·6. In this case the height at ⊗ *m* is 2·1, which, multiplied by 1·6, the product will be 3·4ft. The distance will be set off on the perpendicular C above the L.W.L. at the fore end of the load water line, and a batten from the point so obtained, passing through *m* and D. The height at D and H, and the exact curve of the sheer, will be regulated according to the requirements of the buttock lines and the judgment. The rake and line of the aft side of the stern-post will be then drawn from the deck line, and passing through F. *f* is the aft side of the rudder post.

[Sheering a vessel is a matter of fancy, but a good flowing curve can be constructed as follows: Divide the distance (Fig. 167) *o s* (*s* being the

after end of the quarter timber) into four equal parts, and measure the height at the fore end at *o* of the load water line from the L.W.L. to the deck line. This height will then be multiplied by the following factors at the $\frac{1}{4}$, $\frac{1}{2}$, $\frac{3}{4}$, and *s* points.

	Steamer.	Schooner.	Cutter.
o	1·00	1·00	1·00
$\frac{1}{4}$	·71	·73	·72
$\frac{1}{2}$	·57	·58	·55
$\frac{3}{4}$	·58	·56	·52
Stern end of L.W.L.	·71	·62	·57
s	·81	·72	·72

It will be noted that the rail follows the deck line until near the $\frac{3}{4}$ distance, when it becomes straighter, and causes a graceful taper in the two outlines.]

The stern post, keel outline, fore foot, and stem will next be drawn. The stern post may have great or little rake, according to the judgment. In this case it has been fixed at 35°. If the greatest depth has been fixed, say, at 7ft., set off this distance below the L.W.L. at the point where it has been determined the greatest depth shall be; in this case it has been put at the sternpost. The curve of the fore foot will be determined by the judgment in accordance with the principles set forth in the chapters on lateral resistance and the modern form of stem, as depicted in many of the Plates, that of Isolde being a good example. It is usual for the upper part of the straight stemmed yachts above the L.W.L. to rake forward, otherwise, if the stem is quite perpendicular to the L.W.L., it will appear to rake aft as the vessel sits on the water.

The sheer plan must now be left in order to proceed with the midship section as traced in the body plan.

First produce the vertical line E E (Body Plan 1) at right angles to the L.W.L. This will be the middle line of the body plan.

The *extreme* half-breadth 4ft. must now be set off on the L.W.L. on either side of E E, and through the points perpendiculars will be erected, as represented by *h h*, *i i*.

Next, the depth of the vessel from the L.W.L. to the underside of the keel, and to the lower edge of the rabbet of the keel, at the mid-section are 5ft. and 6ft. 3in. respectively, then set off 5ft. on E E from *k* to *j*; next the width of the keel will be (say) at amidships 9in.; set off this distance *across* E E at *j*. Then divide the distance from *k* to *j* into four equal intervals by the lines or ordinates 1, 2, 3, 4, 5. These lines can then be produced through the sheer plan, and must be drawn strictly parallel to the L.W.L. They will also be numbered 1, 2, 3, 4, 5, if the draughtsman finds it convenient to do so. [The distance from *j* to the L.W.L. can be divided into any number of intervals, but generally 4 will be found sufficient; that number

will give an uneven number of ordinates for the calculation, and moreover, in this case, give an ordinate at the mid-depth, which will be presently referred to.]

The height of the freeboard will next be determined by the rule given on page 19.

$$\frac{\text{Beam} + \sqrt{\text{Length}}}{6 \cdot 8} = 2 \cdot 1\text{ft.}$$

Set off this distance on *i i*, above the L.W.L. at *l*, also on the sheer plan from ⊗ to M.

A rough rule for approximating the area of midsection for any given displacement would be from the rule devised on page 180.

$$\frac{\text{Displacement}}{\text{L.W.L.}} \times 1 \cdot 7.$$

(The displacement will be taken in cubic feet.) This assumes a constant ratio a little fuller than a wave form in the longitudinal growth of the displacement, as explained on page 132, for all vessels, and is only intended as a guide for a trial section. In the case before us we have a displacement of 14 tons, or 490 c.f., and a length L.W.L. of 36. The whole area of misdirection will therefore be

$$\frac{490}{36} \times 1 \cdot 7 = 23 \text{ sq. ft.}$$

The *form* of the curve must be fashioned according to the requirements of the case and the judgment. Having decided on the form, take one of the small splines, and bend it gently to form the curve, as shown in the body plan. The spline will be pinned by weights when the curve appears to be such as required; sweep it lightly in with a pencil, and then determine the area by Simpson's first rule. In this case we have for the lengths of the ordinate:

	Lengths of Ordinates.	Simpson's Multipliers.	Products.
L.W.L.	3·87	1	3·87
W.L. 2	3·40	4	13·60
W.L. 3	2·23	2	4·46
W.L. 4	0·90	4	3·60
W.L. 5	0·30	1	0·30

3)25·83

8·61 × interval of 1·25ft.
1·25

·4305
1·722
8·61

10·7625 sq. ft. = half area.

Thus the half area of the midship section without the underpart of the keel* is 10·76 square feet. The area of the half siding of the keel below No. 5 water-line is ·53 sq. ft., therefore, the total half area of the midsection will be 11·3 sq. ft., and the whole area 22·6 sq. ft. This can be taken as a sufficiently close approximation to obtain the desired displacement. So far the midship section is disposed of, and its second half can be swept in on the left-hand side of the middle line of the body plan. This, however, is a mere matter of fancy, and it is usual to put the midship section on the right hand of the body plan only.

The water line at the mid-depth, between the load water-line and the rabbet of the keel, will now be put in the half breadth plan. Take the half breadth of No. 3 water-line (from the body plan) and set it off on the half-breadth plan on the ordinate at)(to *n*. Next the length of the ordinate at No. 4 station on the half-breadth plan has to be determined. The ordinate)(*n* is 2·23ft., and ·53 can be taken as a good proportion of that length for the ordinate at No. 4. (See page 171.)
$2{\cdot}23 \times 0{\cdot}55 = 1{\cdot}265$ft.

Set off 1·26ft. from 4 to *o* on the half-breadth plan. A moderately stiff batten must now be taken and put over the spots at *n* and *o*. The batten must not be allowed to take a straight line between *n* and *o*, but must be fixed by a weight so as to have a fulness in much the same proportion as the fulness of the L.W.L. In fact, the lines will have almost parallel parts. The line will terminate forward in the rabbet of the stem, but a full explanation of this will be deferred until the sheer plan is complete.

The after body will next be dealt with, and in the first place the mid-buttock line must be determined, but before dealing with the mid-buttock line it will be well to explain what it represents. In the first place on the body plan draw the line *p p* parallel to E E, and in this case at the mid-distance between E E and *h h*. If the model were sawn through in a fore and aft direction, the sectional outline of the parts when separated would form a curve, the outline of which would be shown on the sheer plan, and would represent the run of the quarter; this would be the buttock line, and we will now proceed to trace it. Determine at what point *forward* of F (on the sheer plan) the buttock line shall cut the load water-line; and this distance has been fixed as ·18 of the distance F)(; the latter distance is 16·2ft., and $16{\cdot}2 \times 0{\cdot}18 = 2{\cdot}9$ft. The distance 2·9ft. will be set off on the L.W.L. (on the sheer plan) from F to *q*.

Next the distance *r s* (on the body plan) will be measured from the load water-line at *r* to the point where the buttock line *p p* cuts the

* In practice the keel would be included.

midship section at *s*. This distance is 2·8ft., and must be set off at ⊗ in the sheer plan from ⊗ to S.

The distance *q* to ⊗ has now to be divided into eight equal intervals. The distance is 13·2ft. and $\frac{13 \cdot 2}{8} = 1 \cdot 65$ft., that is, the divisions for the intervals will be 1·65ft. apart, and must be set off perpendicular to the L.W.L. line in the sheer plan at 1, 2, 3, 4, 5, 6, 7.

For the form of the buttock lines a common parabola will be taken, the factors for which are given in the table, page 176. The distance ⊗ *s* will be successively multiplied by these factors, and the products will be the length of the ordinates 1, 2, 3, 4, 5, 6, 7.

8.	....	2·8ft.	×	1·000	=	2·80ft.
7.	....	2·8ft.	×	·984	=	2·75ft.
6.	....	2·8ft.	×	·937	=	2·62ft.
5.	....	2·8ft.	×	·859	=	2·40ft.
4.	....	2·8ft.	×	·756	=	2·11ft.
3.	....	2·8ft.	×	·610	=	1·70ft.
2.	....	2·8ft.	×	·437	=	1·22ft.
1.	....	2·8ft.	×	·234	=	0·65ft.

These distances will be set off on the perpendiculars dropped from the L.W.L. (on the sheer plan) between ⊗ and *q*, and numbered 1, 2, 3, 4, 5, 6, 7. A batten will next be taken and placed over the spots commencing at S, and passing through *q* will intersect the sheer or deck line at a point determined by the height and breadth of the counter. It must be remembered that the drawing is being made *without* the plank, and that the point *q* would be farther aft if the plank were on; in fact, ·17 would probably have been chosen to determine the point.

The buttock line it will be seen cuts the deck at A, and is shown in the half-breadth plan at I. If the deck here were made broader, it would put the point of intersection, I, farther aft, and H would go aft too. This would bring about a contrary flexure in the buttock line in the counter so common a few years ago, but now generally avoided by narrowing the counter aft. This fashion, whilst causing some deck room to be lost, lightens the counter somewhat, a not unimportant consideration in these days of great overhang.

The buttock line has now to be used for determining the shape of the load water-line aft on the half-breadth plan. In the first place, the distance *r k* on the Body Plan must be taken off and transferred to the half breadth plan as *a* E. Through E produce a line parallel to A B; this line will be the buttock line.

Next take the distance F *q* from the sheer plan, and set it off in the half-breadth plan on the buttock line from E to q^1. This will be called *squaring down* from *q* to q^1; that is, if one side of an ⅂ square were placed on the L.W.L. in the sheer plan at the point *q*, the other side would

intersect the buttock line on the half-breadth plan at q^1. The spot at q^1 will be where the load water-line intersects the buttock line on the half-breadth plan, the point here of course being in exactly the same position as the point of intersection in the sheer plan.

Next the ending of the load water-line in the stern-post must be found; the half siding of the stern-post will be drawn at a on A B, as shown. Then a batten will be taken, and starting from the greatest half breadth, at ⊗, will join the stern-post at a, intersecting the buttock line at q^1. Care must be taken that the batten does not spring out beyond the greatest half breadth; nor must the line be needlessly flat, but of such a gentle curve as is consistent with maintaining a good length of body. In yachts, however, which are relatively narrow, the load water-line amidships for some distance is as nearly as possible parallel to the middle line A B.

The water-line No. 2 in the after-part of the half-breadth plan can now be drawn. From the body plan take the half-breadth of the midship section at No. 2 water-line, and transfer it to the line ⊗ on the half-breadth plan, from ⊗ to u. Next, the buttock line in the sheer plan cuts No. 2 water-line at v; from v square down to the half-breadth plan to v^1. Then square down from the aft side of the stern-post (at its intersection with the second water-line) to the half-breadth plan; this will give the extreme stern end of No. 2 water-line. A batten from u, and passing through v^1, will temporarily determine the shape of the second water-line.

As a further aid towards constructing the drawing a diagonal line can be made use of. This diagonal line will be regarded as an approximation to a "dividing line."*

From the intersection of E E and L.W.L. at a in the Body Plan draw the diagonal a b. The diagonal cuts the perpendicular i i and h h at a distance equal to quarter the greatest beam below the L.W.L. It has been found that this diagonal has almost a constant value in existing yachts as follows: Divide the fore body into eight equal intervals and the after

* Dividing line" is the term given by Lord R. Montague to a normal line, running from the stem to the stern, crossing at right angles the vertical projection of each frame. Thus, suppose a vessel to be in frame, and that a narrow plank has been extended round the frames from the stem to the stern; then this plank will take the direction of a *normal line* and rest flat on the frames at points whose surfaces correspond with the plane assumed by the plank during its curvature. The lower edges, or lands, of the planks of boats can, therefore, be regarded as *normal* lines. The term "dividing lines" was applied to them because a *normal* line represents the supposed track any particle of water would take in gliding over the bottom of a vessel. Hence the water is divided, or thrust out of the way, in directions at right angles to the point of contact, so that in fact the *normal lines*, and not the horizontal or water-lines, would indicate the true lines of resistance.

body into four equal intervals. Then the ordinate at each interval will bear the following proportion to the longest ordinate *a g*.

No. 0	0·000	Fore end.
1	0·175	
2	0·350	
3	0·515	
4	0·667	
5	0·800	
6	0·900	
7	0·970	
8 *a g*	1·000	Mid-section.
T	0·960	
U	0·780	
V	0·450	
0	0·000	Stern end.

Divide the base line for the diagonal into eight equal intervals for the fore body, as shown by 1, 2, 3, 4, &c., and quarter intervals for the after body, as shown by T, U, V, 0. Take off the distance *a g* along the diagonal and multiply it successively by the factors just given. The distance in this case is 3·5ft., and this length will be multiplied by the factors as described. For instance, the factor for No. 4 station is 0·667, which multiplied by 3·5 gives 2·33ft. This distance will be set off on No. 4 ordinate from *a* to *e*. (*See* "Base Line" for diagonal, Plate XII.) When the ordinates for the eight intervals and the four intervals have been consecutively computed and set off on their proper stations the curve representing the diagonal or dividing line will be swept in. At the stern end the half thickness of the stern post will have to be allowed for unless a portion of the counter is immersed.

So much of the drawing is now complete that the remaining sections for the body plan can be easily finished. If the draughtsman so chooses he can erase the whole of the ordinates (excepting the one at 𝕸 on the half-breadth plan) and insert new ones at more convenient distances; however, in this case it was not found necessary to remove the ordinates in the fore body, as the divisions were suitable for the working drawing; but the distance *abaft* 𝕸 was divided into spaces equal to the spaces in the fore body, and will be found numbered 9, 10, 11, 12, 13, 14.

A balance section is now taken from the half-breadth plan at number 4 station, and projected in the body plan. Four points in the section are given:

1. Its height above the load water-line on the Sheer Plan at J.

2. The breadth of the load water-line on the half-breadth plan represented by 4 *y*.

3. The half-breadth on the second water-line on the half-breadth plan represented by 4 *o*.

4. The distance $a\ e$ on the projected diagonal line above the Sheer Plan.

For convenience of illustration another body plan has been drawn (see Body Plan 2).

The half-breadth on the deck at J will be so much wider than the half-breadth on the load water-line at y as the case seems to demand; of course, if the bow is to flare out a great deal, the difference will be very considerable; but much flare will produce an ugly shoulder in the immersed line when the yacht is heeled, and should therefore be avoided in racing yachts.

The thin spline batten will now be taken and bent round the spots, taking 4 y, 4 o, and $a\ e$ as transferred to the body plan, and terminate at a point corresponding with the underside of the keel.

The deck half-breadth E to P should now be set off at its proper station on the half-breadth plan (see 4 P), and the deck line can be swept in. Having obtained the half-breadths for the deck line throughout its length, it will be very easy to draw other sections, and it will be best to next fill in Nos. 2 and 6.

The sections of the after part of the vessel can be next filled in, and it will be well to commence with No. 11. There will be six points for this section.

1. The height from the load water-line to the deck, as at K on the sheer plan.

2. The half-breadth on the deck, as at No. 11 L on the half-breadth.

3. The half-breadth 11 y^1 on the L.W.L. of the Half Breadth Plan.

4. The half-breadth 11 x on the Half Breadth Plan for No. 2 water-line.

5. The depth of the buttock line—from the load water-line to z on the Sheer Plan.

6. The distance 11 $a\ i\ i$ on the projected diagonal line.

Having put No. 11 section into the body plan, No. 13 and the others will follow. The water-lines will then be completed, and then it will be time to try the fairness of the drawing by further diagonal lines. A number of diagonal lines will be drawn across the sections in the Body Plan, and then projected as a half-breadth plan (see farther on).

It will be convenient in practice to project the diagonals on the sheer plan above the load water-line or on a half-breadth plan above the Sheer Plan, as shown in Plate XII.

The diagonals started above the L.W.L. will have their fore termination at the point in the rabbet of the stem at the height above the load water-line from which it was projected. The termination of the after end will be

thus found. In Fig. 168 let *a b* be the diagonal in question; from *a* draw *a c* parallel to L.W.L., and cutting the line of the counter in *e*; from *c* drop the perpendicular *c d*, and *d* will be the point where the diagonal line will have its termination abaft any vertical section such as *e e*. If the diagonal be projected *below* the load water-line it will have its termination in the rabbet of the stern-post at the point below the load water-line corresponding with the point in the body from which it was projected. In a similar way the diagonal will have its ending forward at a point in the stem corresponding as to height with the point projection in the body plan.

A batten will be placed over the spot marks obtained from the diagonals on the body plan. [The batten should taper towards the end that has to form the after extremity of the line, as the curve here will be more or less full.] And if the batten takes in all the

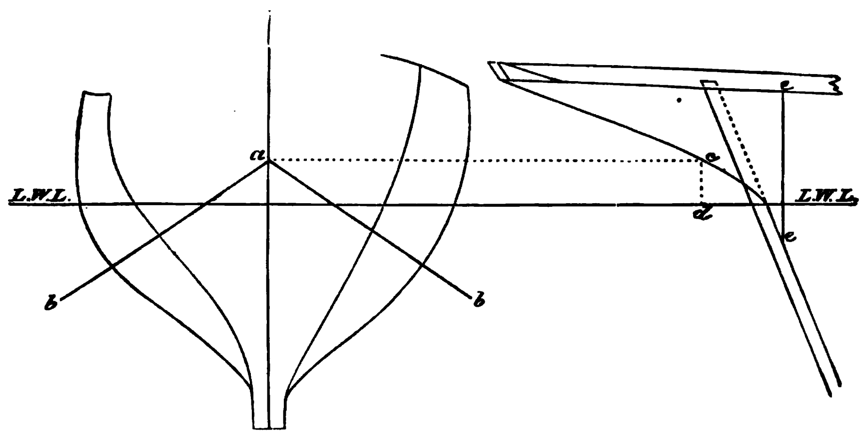


FIG. 168.

spots without any unfair inflections or undulations the line may be considered fair. If, on the other hand, the batten takes in some and omits others, or, by taking the whole in, undergoes distortion, then the batten shows that there is some unfairness in the drawing.

As an additional test of the fairness of the after body another buttock line should be put in the body plan between the one at the quarter breadth and the middle line. This line will be projected on the sheer plan, and must be fair. With all these lines faired the other sections in the body plan can be drawn very rapidly, and when they are complete the body plan will require "fairing" above water. There will, as a matter of certainty, be some unfairness in the after body above water, and to detect it see that two diagonals cross the sections above water in their sharpest curves.

Supposing it is found convenient to start a diagonal from a point on

the middle vertical line E E (Fig. 169) considerably above the height of the vessel's sheer, as at *d*; in such a case, the ending of the diagonal in the counter will be thus found. Measure the height *e a* from the L.W.L. to the end of counter or arch board at the deck; square this distance along from *a* to *b*, next set off the distance *b c* on *e a* as shown by *e x*; and *x* is the point where the diagonal will end aft as projected on the sheer plan. The distance *d o* from the body plan will then be transferred to the sheer plan at No. 15 station, and *d n* at No. 14 from L.W.L. Forward for the ending square is from the top of No. 1 section to the middle line, as from *k m*; then take the distance *s m* and find where it equals a half breadth on the deck line of the half-breadth plan; the point where found will show the distance the diagonal cuts the sheer deck line ahead of No. 1 section. Draw a line at this point on the L.W.L. of sheer plan, and set off *s m* on it. This will be the ending of the diagonal forward.

It will be important, to test the shape of the fore body, to put in an immersed line such as *k l* (Body Plan 1, Plate XII.); it need not, however, be for an inclination greater than 20°. This immersed line must be projected, and if it shows a shoulder or fullness near the entrance, then the shape of the vertical sections above water must be altered until they give such an immersed line as satisfies the judgment of the designer.

It is usual to represent the vertical longitudinal section of the fore body by "bow lines" corresponding with the buttock lines of the after body. Bow lines are, however, owing to their crossing the sections obliquely, of very little value for "fairing." The ending of "bow lines" on the deck line of the sheer plan will be found by squaring up from the intersection of the bow line with the deck line of the half-breadth plan.

Having "faired" the drawing both above and below water, calculation must be resorted to to further test the qualities of the design. First calculate the areas of the various vertical sections; determine thereby the displacement* and the longitudinal position of the centre of buoyancy.

The remaining calculations as set forth in Chapter XIV. will next be made, and the drawing can then be completed for the mould loft.

Lead Keels.

The weight of ballast or lead keel a yacht will carry of course largely depends upon the size and weight of the timber used in her construction

* If the drawing has been made without the plank, a correction will have to be made. The moulded displacement to the displacement with the plank on is as the cube of the moulded beam to the cube of the beam with the plank on. Thus, say the displacement without the plank is 12·5 tons, and that the moulded beam is 7·75ft, the cube of which is 465; the beam with the plank on is 7·9ft., the cube of which is 493; then $\frac{12\cdot5 \times 493}{465} = 13\cdot25$ tons = the displacement with the plank on. Or the displacement due to the plank can be found by multiplying the area of immersed surface by the thickness of the plank.

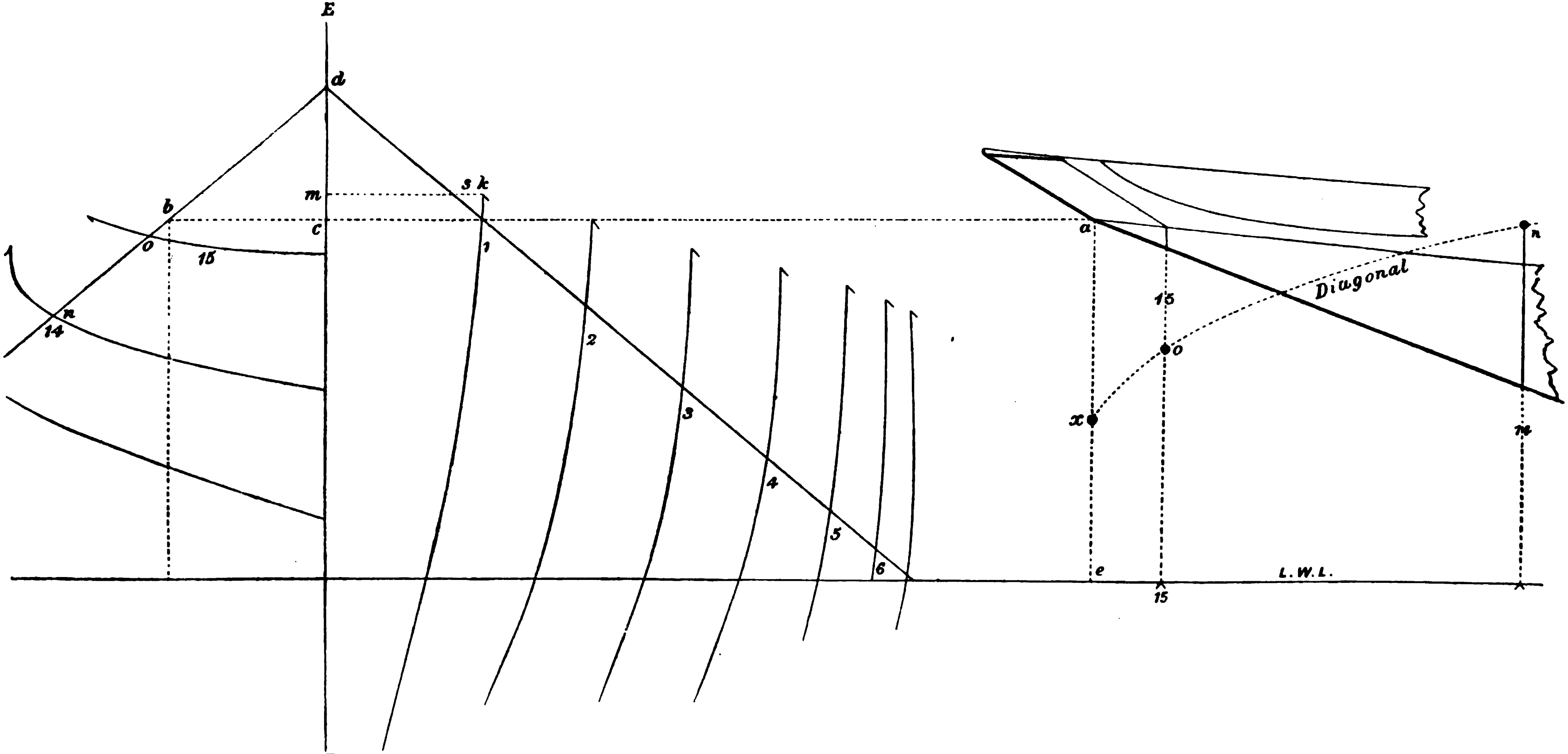


Fig. 169.

and fitting up. A yacht of about 14 tons displacement, built up to the scantlings given in the tables farther on, would carry about 6 tons of ballast, and, of course, all this quantity could be put either inside or outside, or it could be divided.

The shape and weight of the lead keel having been decided upon, its size for that weight must be ascertained; also its centre of gravity in a fore and aft position. Divide the keel into a number of sections, as in Fig. 170. Calculate the areas of these sections, including the two ends 1 and 5. The areas will be summed and the cubical contents formed by Simpson's rule (*see* page 323).

The weight of the lead keel in tons will be found by dividing its cubical contents (as found by Simpson's rule) by 3·16. (There are 3·16 cubic feet of lead to one ton.) The centre of gravity of the keel will be calculated from No. 1 station (Fig. 170) by Simpson's rule (*see* page 323).

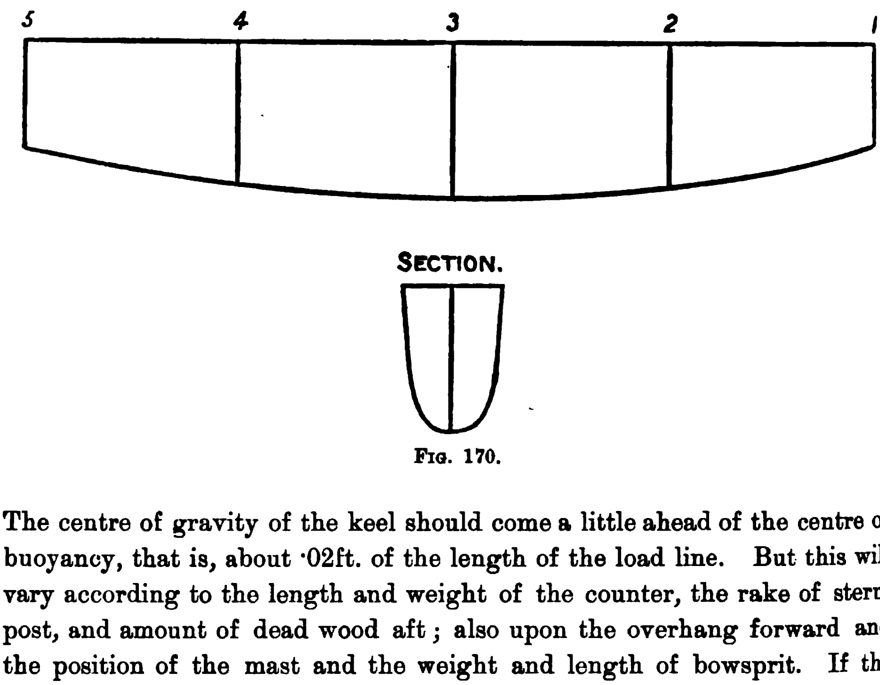


FIG. 170.

The centre of gravity of the keel should come a little ahead of the centre of buoyancy, that is, about ·02ft. of the length of the load line. But this will vary according to the length and weight of the counter, the rake of sternpost, and amount of dead wood aft; also upon the overhang forward and the position of the mast and the weight and length of bowsprit. If the whole of the lead is to be put on the keel, these matters will have to be well considered.

In the case of a "fin bulb" keel the weight and centre of gravity of the bulb would be calculated by the same process; and the weight of the fin from the table of weights of metal plates and its centre of gravity by the rules for figures as described on page 326.

In making moulds for the lead keel $\frac{1}{8}$in. per foot each way will have to

be allowed for shrinkage. In purchasing the lead—especially if it is old lead—1cwt. per ton should be allowed for dross.

The Making of Models.

Happily the system of making the model first, and then taking the lines from it, has been banished from most yacht building yards, and models are now seldom used excepting for the purpose of experiment, or as part of the furniture of a museum.

The simplest plan to pursue in making a model will be to trace all the water-lines, and then transfer such tracings to clean pieces of deal of a thickness to represent the distance the water lines are apart on the scale.

The tracing will include the whole breadth, and not the half-breadth, if the model be for experiments. The form of the proper water-line should be traced on both planes that come together—that is, the plane representing the L.W.L. should have on its under side the tracing of water-line No. 2; and No. 2 should have on its top side a tracing of its own plane, and on its under side a tracing of No. 3, and so on.

One or two planes *above* the load water-line will require to be projected; these can be easily taken off from the body plan. Their form aft in the counter will probably be a very round curve, and a point in the curve will be obtained by squaring down from the buttock line on the sheer plan to the buttock line on half-breadth plan. Of course the line representing the plane must be drawn in the sheer plan above and parallel to the load water-line. The ending of the plane aft will be found by squaring down from the counter at its intersection with the line that is to be projected.

When the lines have been traced on the surfaces of the wood the edges can be trimmed off to near the tracing lines. The pieces will be then screwed together, and the model, by aid of the spokeshave, &c., cleaned off. The model can very easily be afterwards hollowed by unscrewing its parts, and sawing out as much of the middle part as may seem necessary.

A model can be made without reference to lines by screwing thicknesses of wood together to represent a block, and then shaping the latter into the form of a vessel; the lines can afterwards be easily traced with a pencil, by unscrewing the parts, and laying them on the drawing paper.

CHAPTER XVI.

LAYING OFF AND TAKING OFF.

Laying Off.

Laying off a vessel on the floor of the mould loft is simply the process of transferring and enlarging to full size the drawing made on paper. There are many good treatises on this subject, and one by Mr. Thearle can be specially recommended to those who desire to become acquainted with all the details of laying off.

The instruments required for laying off will comprise measuring rod and rule; two squares, a bevel; a chalk line; compasses; a straight edge; and fir and American elm battens.

The fir battens should be $1\frac{3}{4}$in. by 1in., and from 40 to 60 feet in length, according to the length of the vessel. These will be used for the sheer line, the water lines, and the diagonal lines. The American elm battens will be $\frac{1}{2}$ inch by $\frac{3}{8}$, tapering at one end, and from 10 to 20 feet in length. These will be required for the body plan, rudder, forefoot, &c. The chalk line will be required for making straight lines of great length, such as the load water-line in the sheer plan.

If American elm battens cannot be had, some made of pitch pine, or any red pine, will do almost equally well. The battens are held in position by finely-pointed iron pins, about 4in. long; or nails will of course do. The pins or nails are not put *through* the batten, but one is placed on either side so as to hold the batten at the "spots."

In the first place, the sheer plan must be delineated on the floor, showing the deck or sheer line, and the outer edge of the stem and stern post, and lower edge of the keel. Distances will be set off on the load water-line, to correspond with the stations for the vertical frames. Then, at a convenient distance below the load water-line, a line will be drawn to represent the base line of the half breadth plan. Stations for the cross sections will also be set off on this line, and from these stations draw lines to meet the similar stations on the load water-line of the sheer plan; these lines can be made with the chalk line.

It will be assumed that the sections in the body plan have been drawn without the plank.* Then across the plan draw a number of diagonal lines, as shown in Plate XII. and Fig. 173. When there is a sharp curve to the bilge, and when the character of this curve is apparent above water in the counter, great care must be taken to get a diagonal as nearly as possible through the middle part of the curve, as shown in each section, for the purposes of fairing. Having drawn the diagonals on the body plan at suitable intervals, the projections of these lines will be put on the half breadth plan or sheer plan, as may be most convenient.

First, however, project on the half breadth plan the deck line less the plank, and the load water-line less the plank.†

The diagonal distances from the middle line of the body plan to each frame will be measured, and set off on their proper stations on the half breadth plan; a curve will be swept in through the points thus obtained, and this curve is termed the "rebatement" of the diagonal, or its real shape. The ending of the diagonal line in the rabbet will be found by a process similar to that used for the ending of the water-lines. The batten will be kept in position by iron pins or nails on either side of the spots, and when the line is "faired" it will be drawn to the batten by a thin piece of chalk. No battens, whether those of lancewood, elm, or fir, should be kept pinned round curves longer than is necessary.

* In cases where the sections in the body plan have been drawn to the outside of plank, the thickness of the plank should be taken off before the drawing is passed into the mould loft. A convenient method of removing the plank will be by drawing a number of semi-circles, or arcs, such as *a b c* in the annexed cut, Fig. 171; for radius open the compasses equal to the thickness of the plank; place one point on the section A B and describe the arc; then take the batten or mould and draw a line touching the back of each arc as shown by the inner line *o k*. The point *o* will be the middle of the rabbet in the keel. Some builders allow for the thickness of the plank as they lay off from the diagonals, or put a line in the drawing next to the middle vertical line of the body plan, equal to the thickness of the plank, and measure from that. The plank can be put *on* a vessel by an analogous process.

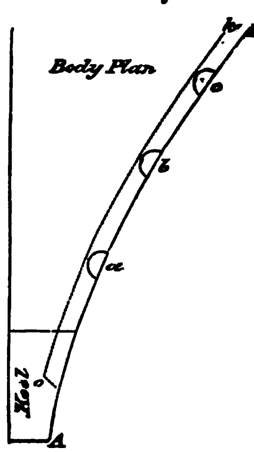


Fig. 171.

† To find the ending of the load water-line draw a horizontal section of the half siding of the stem as *a b c*, Fig. 172; from *c* inwards set off the thickness of the plank; the point *d*, the middle of the rabbet, will be the ending of the water-line; *s* will be the bearding line of the rabbet; *d* the middle line, and *c* the outer edge of the rabbet—the one usually shown on the sheer plan.

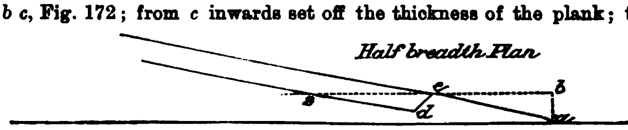


Fig. 172.

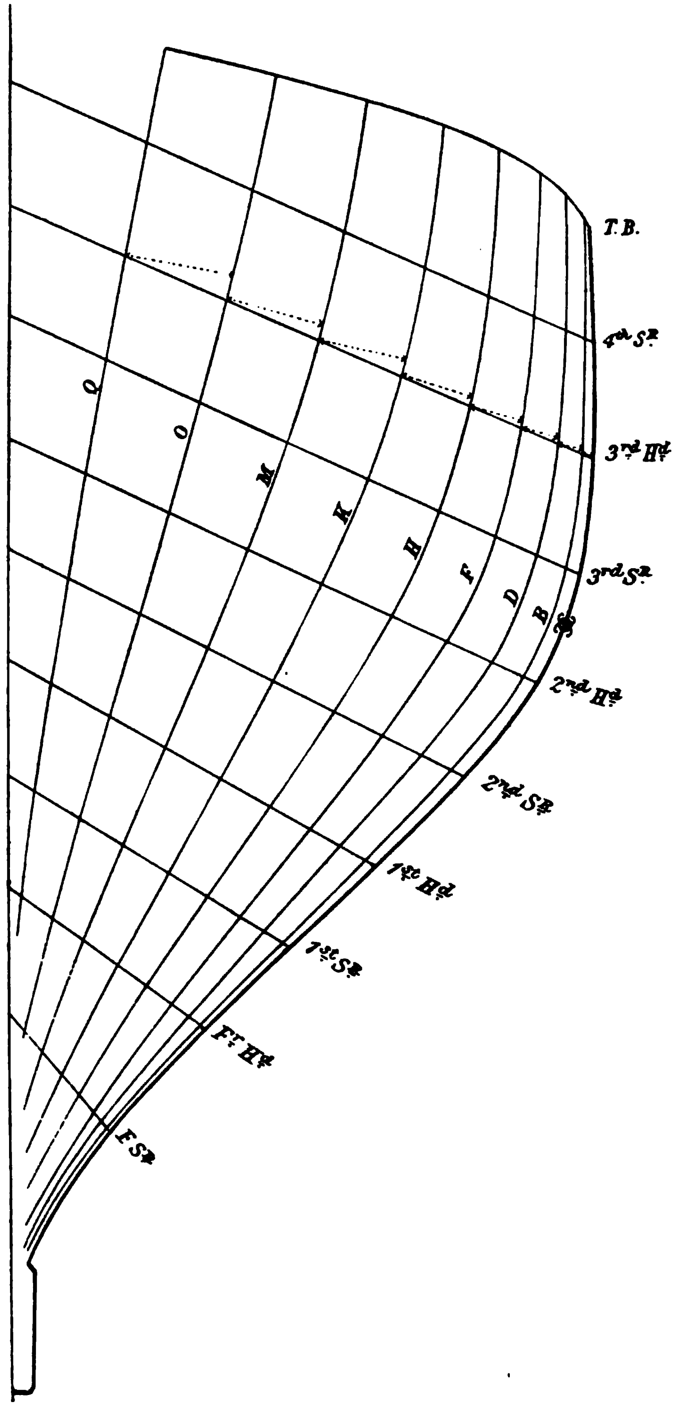

T.B.
4th Sn
3rd Hd
3rd Sn
2nd Hd
2nd Sn
1st Hd
1st Sn
Ft Hd
F Sn
Q
O
M
K
H
F
D
B

Fig. 173.

When the whole of the diagonals have been drawn on the half breadth plan, the vertical sections will be constructed by transferring the half breadths to their proper diagonals on the body plan. For this purpose a plan must be drawn on the floor agreeing in every respect with the drawing on paper. When the body plan is complete the moulds for the frames can be made. The diagonals must be marked on the moulds as well as the L.W.L.

It is usual in large vessels that are very full at the bow and stern to "cant" the frames in the fore and after body, so as to keep them as nearly as possible square to the horizontal lines. Canting the frames is now, however, not always practised in framing a sharp vessel like a yacht; but the bevellings must be carefully observed.

If the floor is not large enough to admit of the drawing being laid off in one piece, the sheer and half breadth plans will be divided as required. In such cases two at least of the same sections must appear in each part or division; otherwise there will be a want of continuity or fairness in the lines.

Before describing a method of obtaining the bevels for the frames of a yacht, which, as before said, are now seldom canted, it will be understood why the bevels are required. On Plate XII. refer to the Half-breadth Plan, on which the diagonal is projected. At B is shown the section of one part of a frame, and it will be seen that the frame is not "square," but bevelled to suit the angle of the diagonal. To obtain these bevels the following method is in use by most yacht builders: On the body plan on the floor of the mould loft draw in the diagonals Fr. Hd. (Floor Head), Fig. 173, 1st Hd., 2nd Hd., &c. (T B is top breadth). These will represent the parts of the frames. Then draw in the intermediate diagonals, F. Sk. (Floor Sirmark), 1st Sk., &c. We will select the 3rd Head as an illustration. Take a thin piece of board, or strip of drawing paper, and lay it along the diagonal (see Fig. 173), then mark on it the distance on the diagonal from the mid-section ⊕ to the next section B.*

Then shift the paper until the mid-section mark comes on B section; then mark the distance D section is from B. Then shift the paper till the mid-section mark comes on D section; then mark the distance F is from D, and so on all the way up the diagonal, but always measuring the distance at right angles, as shown at the 3rd Head by the arrow heads.

Next prepare a bench or table, the top of which is shown by A A, Fig. 174. On the edge of the bench screw a piece of board B B of the thickness of the bevel board. Draw the line C D at right angles to B B.

* B. It should be noted that in the mould loft the sections in the fore body are always lettered A, B, C, D, &c., and numbered in the after body; A and No. 1 come next the mid-section. In the example before us every other section is omitted.

E is a bradawl or nail in the table to rest a straight edge against. The distance of E from the inner edge of the board B B will be the distance the sections are apart in a fore and aft direction, and it must be noted that in the mould loft no sections are omitted from the body plan.

Take the strip of paper and transfer the marks thereon to the inner edge of the board B B as shown at *x*. Then fix the bevel board F F close to B B. Take the straight edge and press one edge against the nail at E, and the other on the mid-section mark, and draw a line across the bevel board. Then shift the straight edge to the B mark, and so on, until all the bevels have been marked as shown on Fig. 175.

The board is now placed in the hands of the top sawyer who it will be

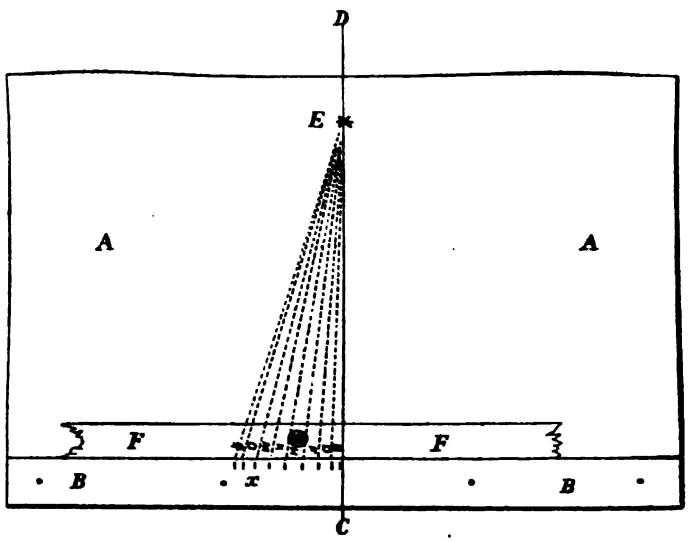


Fig. 174.

assumed has by aid of the mould marked out the frame on a suitable piece of timber. See H H, Fig. 175.

The underside of the timber will now have to be marked to guide the bottom sawyer, and it will differ from the outline of the frame marked on the top according to the bevels.

Take the bevel and get the proper angle from the bevel board. Then put it on the timber at its proper diagonal, as shown on K and on H. With a rule or piece of wood measure the distance *a* to *o* on the edge of the mould H; then transfer the distance to *s g*, and *g* will be a mark for drawing in the curve of the frame on the underside of the timber. Other marks in the curve will be similarly obtained. Occasionally the bevels are not sawn, but set off and adzed away after the timber has been cut out in square form.

TAKING OFF.

If a vessel could be put ashore just as she floats, that is, in such a way that her load water-line would remain in the plane of the horizon, the work of taking off would be very simple. A square, having both sides marked in feet and inches, would be placed at the stations where it was required to measure the sections (see Body Plan, Plate XIII.). The distances from the perpendicular, or upright arms of the square, to the vessel's side would then be measured in a horizontal direction; and the distances from the *horizontal* of the square to the bottom of the vessel would be measured

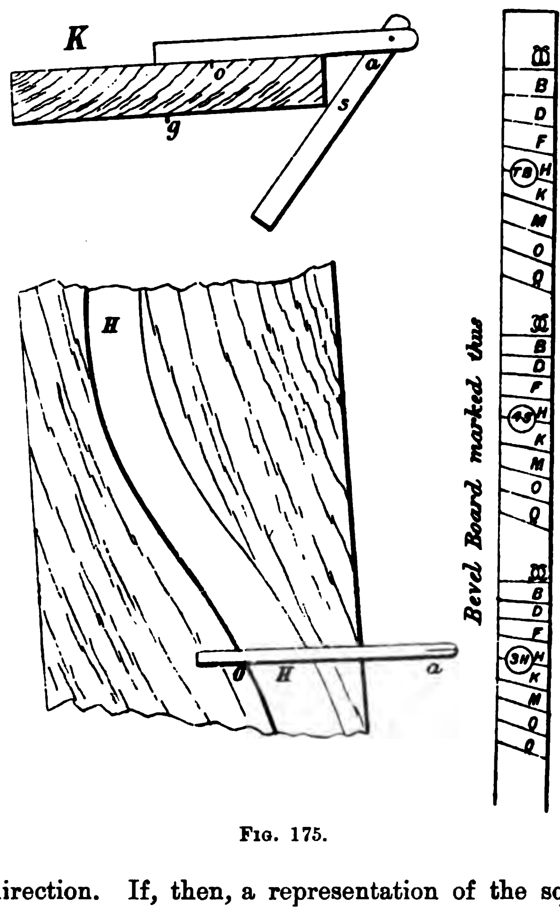


FIG. 175.

in a vertical direction. If, then, a representation of the square, made to some scale, be drawn on paper, and the measured distances properly set off, points would be obtained for making the curves of the various sections.

However, as there are so few conveniences for docking a vessel, it is not often that one can be placed in the position we have described. A vessel when laid ashore or hauled up on a slip is usually considerably down by the stern or head, and to "take off" a vessel so placed from

squares erected so that the load water-line should still be the base of the operations would be a work of great tedium and difficulty. To avoid this difficulty the load water-line is disregarded so far as it being used as a base line is concerned, and a perfectly *horizontal* base line is used instead. This base line is placed at the keel, and can be set off by the aid of a spirit level and chalk line or straight edge. If a vessel is so much down by the stern or head that a horizontal base line cannot be obtained to run the whole length of the vessel (see Base Line, No. 1, Plate XIII.), it must be divided into two or more parts. In the example here given of taking off the Cygnet, a second base line (see Base Line, No. 2, Plate XIII.) only was required, and unless the vessel be very much indeed by the stern or head, or shows very little depth of keel, two base lines will generally run the whole length of the vessel. The base line should be snapped across the rudder and terminate at the fore foot, as shown in the example.

Having obtained the base line, the length of the vessel on deck will be measured from the fore side of the stem on deck to the half side of the stern-post. This length must not be measured close to the deck, or there will be an error, owing to the deck at the bow being so much higher out of the water than at stern. This error can be avoided by keeping the tape or rod as nearly as possible on a line parallel with the load line, and finally squaring down to the back of the stern-post for the length. In the case of the Cygnet, however, the error was very small, and was thus found : The length measured *along* the deck was 54ft. 10in. = 54·83ft., which, multiplied by the cosine (0·99869) of the angle made by the deck with the load water-line, gives 54·75ft. Thus the error in length would be only 1 inch. The cosine will be most readily found thus: Divide the length on deck by the difference in the height of the deck at stem and stern measured from the load water-line. In the case of the Cygnet the difference is 2ft. 9in., then $\frac{2 \cdot 75}{54.83} = \cdot 05$, the sine of 2° 56′, found from a table of sines. The angle being thus known, it is easy to find the cosine from a table. In practice, unless the vessel has a very tremendous sheer forward, or is very long, the length measured on deck, as nearly as possible in a direction parallel to the load water-line, and squared down to the back of the stern-post, will be sufficiently accurate.

The length of the vessel having been determined, her various half breadths must be measured on deck *square*, that is, at right angles to the middle longitudinal line of the vessel. These half breadths can be measured at equidistant intervals, but this is not necessary. For instance, in the example now before us the half breadths were taken at irregular intervals (see Sheer Plan, Plate XIII.). Thus No. 1 is 5ft. 11in. from the fore side of the stem on deck; No. 2 is 5ft. 10in. from No. 1; No. 3 is

5ft. 9in. from No. 2; No. 4 is 6ft. 9in. from No. 3; No. 5 is 6ft. 8in. from No. 4; No. 6 is 7ft. 8in. from No. 5; No. 7 is 6ft. from No. 6; No. 8 is 6ft. from No. 7; and No. 9 is 4ft. from No. 8, and comes at the back of the stern-post. These distances added together will be found equal to the length on deck.

It is always best to measure the *whole* breadth on deck, and halve it when the drawing is being made.

It may be convenient (if the middle fore and aft line of the vessel cannot be very easily determined on deck) to take the breadths across the vessel from stanchion to stanchion, or across one of the ends of a skylight; this will insure that the breadths are taken *square* to the middle line of the vessel. [The longitudinal distances between the breadths should be measured as nearly as possible parallel to the load water-line.]

As the breadths are measured on deck mark on the outside of the vessel on the covering board the *exact point* where each breadth was taken. This will indicate the stations for measuring the shapes of the sections.

Next drop a plumb from the fore side of the stem on deck to cut the base line (which can be extended beyond the fore foot by a straight edge) at A (see Sheer Plan, Plate XIII.). From this plumb line set off, along the base lines, the *interval* between each breadth as measured on deck. This will be done by plumbing down from the deck or covering board and squaring into the keel. If the vessel is nearly as she floats in the water, the distances can be set off on the base line by a rule or measuring rod, starting from the plumb A. These stations on the base lines will correspond with the stations for breadths previously set off on the covering board.

The load water-line will now be taken. Ascertain where the load water-line is marked on the stem at z, and plumb down to No. 2 base line by the plumb at C. Measure the length of the plumb line from the base line to z or L.W.L.: this will determine the height of the L.W.L. above No. 2 base line. The height of the L.W.L. above No. 1 base line at the stern-post will be found in a similar way by the plumb line at D x, and measure the distance a, a^2, and a^2, x.

For rake of stern-post measure along No. 1 base line the distance from No. 8 station at a to where the base line cuts the stern-post at b. As it is known where the stern-post shows on deck abaft No. 8 station, its rake can easily be determined.

The shape of the rudder will be found by taking a series of breadths in any way that may seem most convenient.

The rounding up of the fore foot will be found by measuring (at certain intervals) the horizontal distances from the plumb line A to the stem. Care must be taken that these measurements are taken at right

angles to the plumb line A; and the distance from the plumb line A to the point where the load water-line cuts the stem at *z* must be accurately measured.

The depths *below* the base lines to the under side of the keel have to be measured at each station; then when the heights above the base lines to the covering board at each station are known, all necessary measurements for the sheer plan have been taken. The height will be taken in the course of taking off the sections.

To take off the sections a square marked in feet and inches must be put up with its lower limb at *right angles to the keel* at each station marked on the base lines, and will appear as shown (Body Plan, Plate XIII.). Next a plumb line (E) is dropped over from the deck at one of the stations. The point where this plumb line cuts the horizontal limb of the square at *c* must be accurately observed and the distance measured from the keel of the vessel at *f* to *c*.

This plumb line (E) will also "prove" the upright arm of the square. The horizontal distances *d d d d d* will be measured from the *plumb line* to the side of the vessel: care must be taken that these distances are taken *at right angles to the keel.* This can be ensured by plumbing to the lower limb of the square. Any *tumble home* must be measured if the plumb line touches the *side* of the vessel and not the *deck edge.* Before removing the plumb line E measure the *height from the base line to the deck* at each station.

Next take the plumb line and at certain intervals plumb from the vessel's skin to the lower or horizontal limb of the square. See *e e e e e* (Body Plan, Plate XIII.) and measure the length of the plumb line very carefully in each case. The intervals *e e*, &c., need not be equal; but the exact interval must be accurately measured.

Having measured the various sections at the stations, nothing remains to be done but to measure the height of the bulwarks; the length of counter abaft the stern-post; and the height of the arch board of counter above No. 1 load line. This latter measurement will be taken by a plumb line as shown on the sheer drawing at F. The height of the deck above the base line at the stern post will be found by the plumb line at B.

It will be found convenient, and will save labour, if, instead of an ordinary line for the plumb, a piece of measuring tape, twelve or fifteen feet in length, be used. All the distances plumbed can then be accurately read off at once without having recourse to a rod or rule.

To put the measurements on paper in the form of a drawing the following process will be observed:

The Sheer Plan will be first drawn: Draw lines to represent the

base lines No. 1 and No. 2. On these lines set off the stations, 1, 2, 3, 4, 5, 6, 7, 8, 9, and the length taken on deck. The load water-line will next be put in; on the base line set off the distance from a to a^2, and from b to a^2; next erect the perpendicular a^2, x; through x produce x b, this will be the stern post, and x will be the point where the stern post is cut by the load water-line. At the fore end the length of the plumb line C will be set up from the base line to z; then the load water-line will be drawn through x to z. The various heights *above* the base lines as taken for each section by the plumb line E (see Body Plan) will be set up from the stations 1, 2, 3, 4, 5, 6, 7, 8, 9; the sheer or deck line can then be swept in. Next, the depths *below* the base line will be set off, and the line of the keel drawn. The manner of completing the other parts of the sheer plan requires no explanation.

For the body plan draw lines to represent the "square" and the points on it where the measurements were taken. Set off from g on the lower or horizontal limb of the square, the *exact half breadth of the deck* measured from the middle line H H. To this half breadth add the amount of tumble home, if any; this will give the point c on the square. From c towards g set off the distance c f, and f g will of course be the half thickness of the keel, f being the point in the keel against which the end of the square rests.

The intervals e, e, e, e, d, d, d, d, will then be set off; and lines drawn from such points at right angles to the sides of the square. On these lines set off the distances measured with the plumb and rule. Nothing remains but to sweep in the shape of the section.

The whole of the vertical section being completed the water lines will next be drawn. On the sheer plan at such intervals as may be decided upon, draw the water lines W. L. 2; W. L. 3; W. L. 4; W. L. 5.

Take the compasses, and on the sheer plan at No. 1 station, base line No. 2, measure the distance from h to i; on the body plan, base line No. 2, set off this distance from h to i, *square* to the base line. Next proceed with No. 2 section measuring from j to k, and setting off the distance on the body plan from j to k. For No. 3 section the distance l m will be taken from the sheer plan and set off on the body plan; for No. 4 section the distance c o, measured from *base line* No. 1.

Through the points i k m o in the body plan draw the curved line L.W.L.; this will be the water-line of the fore body. The other water lines will be obtained in a similar manner.

The base line of the half breadth plan (I J) is drawn *parallel* to the base lines on the sheer plan, and the lines (1, 2, 3, 4, 5, 6, 7, 8) representing the sections are drawn at right angles to these base lines.

From the body plan, at right angles to HH, take off the half-breadths *p i, p k, p m, p o,* and set them off on the half-breadth plan at *p i, p k, p m,* and *p o,* and so on for the after body. The half-breadths taken *below* the load water-line, as at *p q,* will represent the lower water-lines on the half-breadth plan. Thus all the water-lines can be swept in.

When the water-lines and deck-line are completed in the half-breadth plan, a new body plan can be made with the L.W.L. as the base line. First determine the distances the new sections shall be apart on the load water-line (L.W.L.) of the sheer plan at K, L, M, ⊗, N, O, P, Q. At these stations draw lines at *right angles* to the L.W.L., as shown by the ticked lines.

Next, on the base line of the half-breadth plan, set off the correspond-

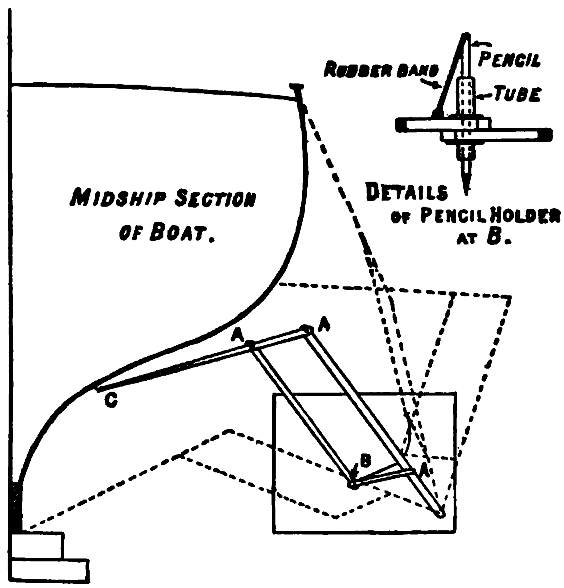


Fig. 176.—Dotted lines represent centres of pantograph at different parts of section.

ing stations at K, L, M, ⊗, N, O, P, Q; from these points draw ordinates at *right angles to the load water-line as represented on the sheer plan.* Then the half-breadths for the new body plan will be thus found. At ⊗ (or any other station) measure the half-breadths from ⊗ to *r, s, t, u,* and *v, square,* or at right angles to the base line I J; these distances will be the half-breadths for the new body plan which, when complete, will appear as that of the Kriemhilda, depicted in Plate I.

Mr. Cecil A. Allen, of Bangor, co. Down, in 1891 proposed that it would be simpler to take off the lines of a yacht in sections by aid of a pantograph instead of by the usual shipbuilders' method. The pantograph was made for taking off the sections of a 26ft. yacht as follows: Arms, 3ft.,

and arranged to reduce to a scale of 3in. to the foot. The joints at A (Fig. 176) were made with ¼in. iron bolts; a short metal tube forms the joint at B, and makes a convenient holder for the pencil, the latter being free to slide in the tube, and kept to the paper by means of a light rubber band. A base line must be struck on the keel of the boat parallel to the water line, and on this the sections must be spaced off every 2ft. or more, as may be convenient, from the bow. The same on a line stretched from stem to stern on deck; these points are then squared on to the rail, care being taken to have the marks on the rail and base line square above each other; the section lines were then chalked down the skin of the boat.

A drawing board, with pantograph attached, is placed in a vertical position at right angles to the fore and aft centre line, and opposite the section to be traced, at such a distance that the arm C could reach all parts of the section. The point of C is then traced down the chalk line, and a reduced copy of the section is obtained on the paper at B; all the important points, such as base line, top and bottom of keel and gunwale, being marked with a dot, which was done by pressing the top of the pencil when the arm C was opposite any of these marks. The half breadth at rail and of keel at base line are then measured and noted on the section; the remainder of the sections are done in a like manner.

The curve of the stem bow and the ornamental design on the head rail can be taken, also the outline of the rudder, this being done by placing the pantograph in a fore and aft line with the boat.

In making out the drawing of the boat, a sheet of tracing paper should be pinned on the board, and marked with a vertical centre and horizontal base line, the sections obtained by pantograph being slipped under and traced when the centre and base line of the sections are right with those on the tracing paper; the sheer heights are then taken from the base line, and sheer plan, &c., made out in the usual way.

By this method the lines can be taken off equally as well when the boat is lying on her bilge as when she is upright.

Taking off Lines from a Block Model.

Measure half-breadths on deck at certain stations, and the *perpendicular* heights from the load line to the deck. Set off these distances on paper. Then take a strip of lead of a convenient length, and in size about ¼in. by ⅛in., and fit it to the model at each station. Mark the load water-line on the strip, and the deck-line; then carefully lift the strip

from the model, and place it on the paper so that the marks on it cut the load water-line and the half-breadth line of the deck at its proper point. The tracing of the section will be completed by running a pencil round the curve formed by the strip.

An ingenious instrument (the pantograph) is manufactured by Mr. Stanley, Great Turnstile, London, for taking off lines of models; and Mr. G. L. Watson, the well known naval architect, has invented an instrument of a similar character for taking off the lines of a model.

CHAPTER XVII.

BUILDING.

THE keel and floor construction of a yacht will be first described, as the operation of building commences therewith.

Fig. 177 illustrates the old-fashioned orthodox structure of a vessel's

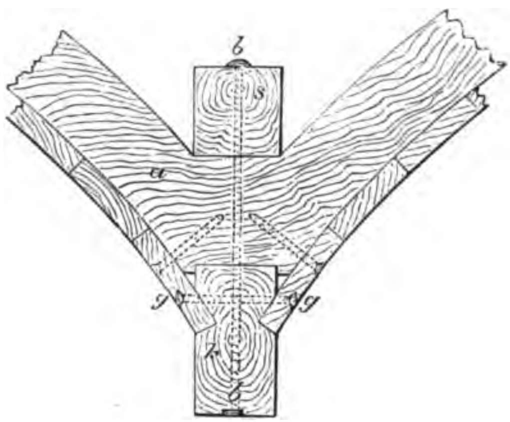


FIG. 177.

floor: *a* is a grown floor, sometimes jogged over the keel as shown, and sometimes not, to the main keel *k*. The keelson is shown by *s* and garboard by *g*, bolted through the keel. The keelson, floor and keel are bound together by through bolts, as shown by *b*; but we have come across vessels in which the keelson has only been secured to the floors by diagonally driven bolts or spikes. The strength of these wood floors is mainly dependent upon the grain in the throat having a natural bend into the arms of the floors. If the grain in the throat is vertical, the chances are that the through keel bolt would split the floor; or if the grain runs across the arm, as shown at *a*, there will be little strength; but it is very rarely that a floor which has not a naturally grown crook is found in a yacht.

Messrs. Camper and Nicholson introduced cast-iron floors of the form shown by *a* (Fig. 177), upon the heads of which the heels of the timbers

above are dowelled, and the heels of the sister timbers are brought down to, and rest on, the top of the keel; Mr. John Harvey elaborated this plan by lead floors cast with angle iron in the inside. Either of these plans were excellent, both for strength or bringing about a low situation of the centre of gravity, but the practice of putting the whole of the ballast outside has rendered this plan of floor construction obsolete.

An approved way of constructing the floor and keel of a wood yacht was introduced in the year 1858, and has since been largely used. Fig. 178 is an illustration of this plan, and was carried out in many

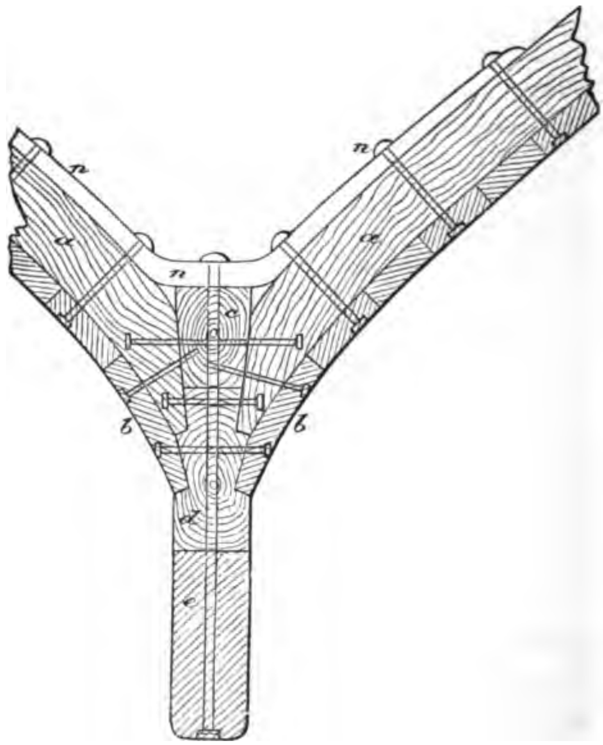


FIG. 178.

yachts. It will be seen that the grown floors are dispensed with, and that the heels of the timbers *a a* are rabbetted into the keel *d* and keelson *c*, the latter being also termed a "hogging" piece. The heels of the timbers are bolted through keel and hogging piece. The frames or timbers are further secured to the hogging piece and keel by the iron knee floors *n n*, and the throat of the knee floor carries a long bolt through hogging piece, keel, and lead keel *e*. But this is not a good plan, and the most approved practice now is to have bolts through the wood keel only for the iron knees, and

separate bolts for the lead keel set up on the top of the wood keel by nuts. The garboard strakes, *b*, are bolted through the main keel, and altogether the structure is made as solid and immovable as if it were one

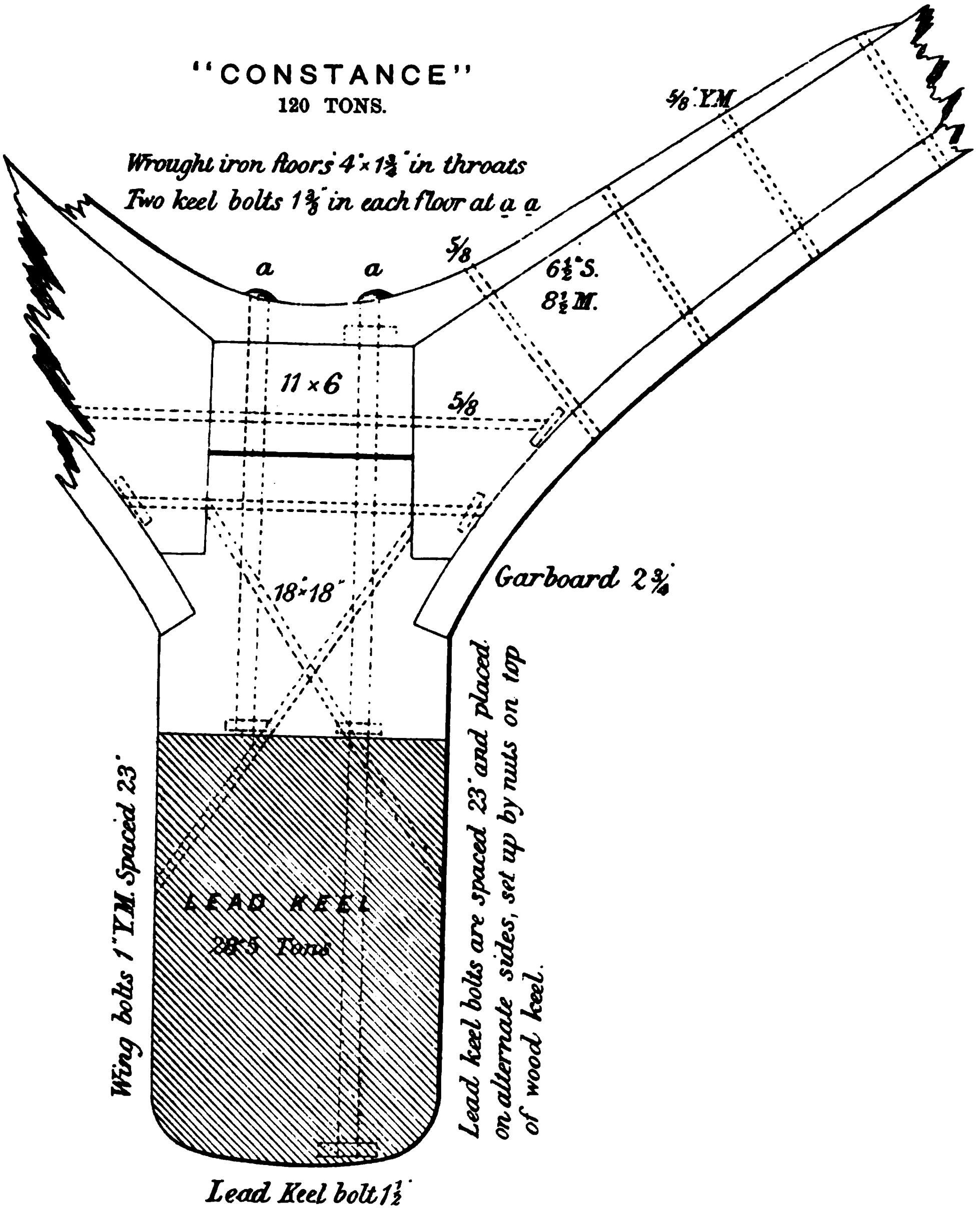


Fig. 179.

whole piece. In building according to this plan great care is necessary in fitting the heels of the timbers into the keel and hogging piece, and in seeing that they are of sufficient depth so that the bolts may be well

placed. A later adaptation of this form of construction is shown on Fig. 179, representing the floors and keel of the yawl Constance (now Freda), built in 1885.

Sometimes the siding of the hogging piece is a little smaller than that of the main keel, and the heels of the timbers are made to rest on the edges of the top of the keel, which thus form the stepping line on either side.

The plan of bolting the heels of timbers to the sides of a keel without any jogs or stepping line whatever is very objectionable, because there is a probability, in case a vessel got ashore and bumped heavily,

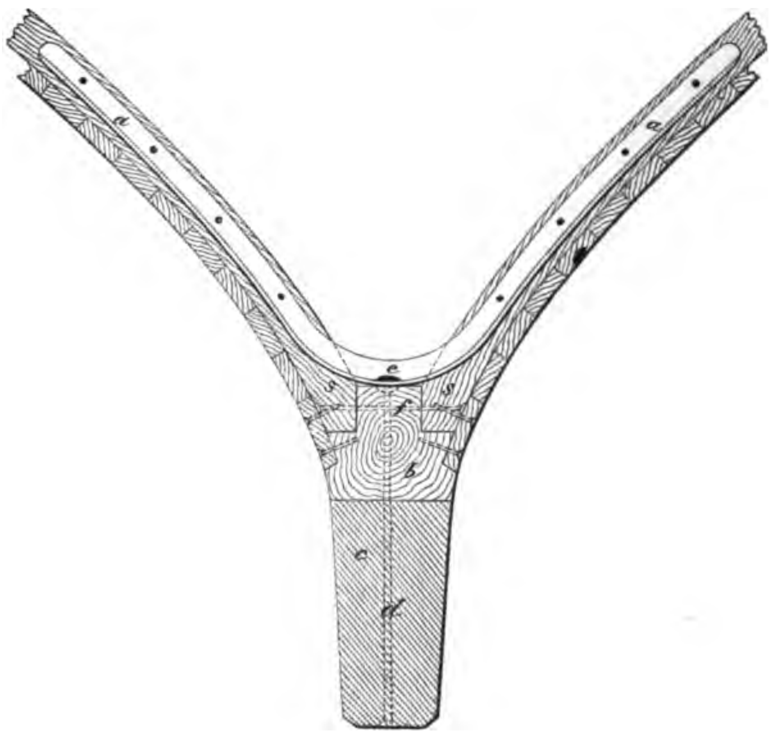


FIG. 180.

of the garboards being driven out of the rabbets. In a vessel with a flat floor this assuredly would be the case, as the iron knees would most likely part in their angles at the first heavy bump; in fact, they would have to bear the brunt of the shock. But even with a great rise of floor, the garboards may be crushed out of the rabbets, if the heels of the timbers are not butted into the keel; in fact, a case occurred in 1877 when a vessel of some 120 tons weight got on some rocks in Belfast Lough, and her garboard strakes were split and forced out of the rabbets. The smallest movement of the heels of timbers down the side of the keel would serve to wrench the garboards away from their fastenings or

crush them; whereas, if the heels were properly butted into the keel they could not shift, and no undue strain would come on the garboards in case of bumping.

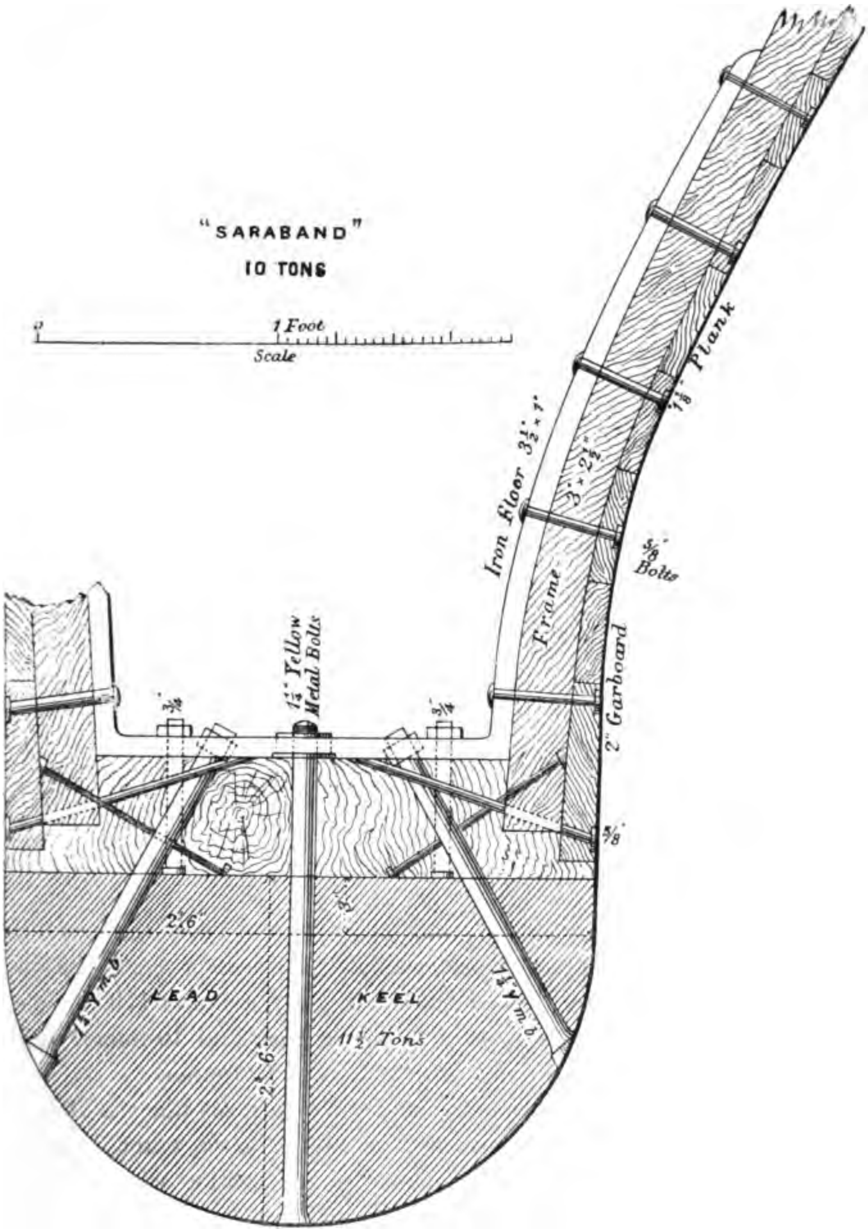


Fig. 181.

The late Mr. Dan Hatcher, of Northam, on the Itchen, for many years used a plan for fixing floors which can be recommended for small yachts.

In Fig. 180 is the keel, about 8 inches sided amidships for a 10-tonner of the past. A stepping channel, about 4 inches deep, is cut out along its entire length for the heels of the timbers *s s* to rest on, and a bolt *f* passes through the heel of each timber and the keel. Besides this an L angle iron knee, *a*, is bolted on the sides of the timbers through the keel, as shown. The through bolt *d e* passes through the keel and lead keel; but, as before explained, it should not pass through the angle iron knee floor. The latter should have separate and smaller bolts. In large yachts a through bolt is driven through garboard and keel, and through garboard and a timber in every frame.

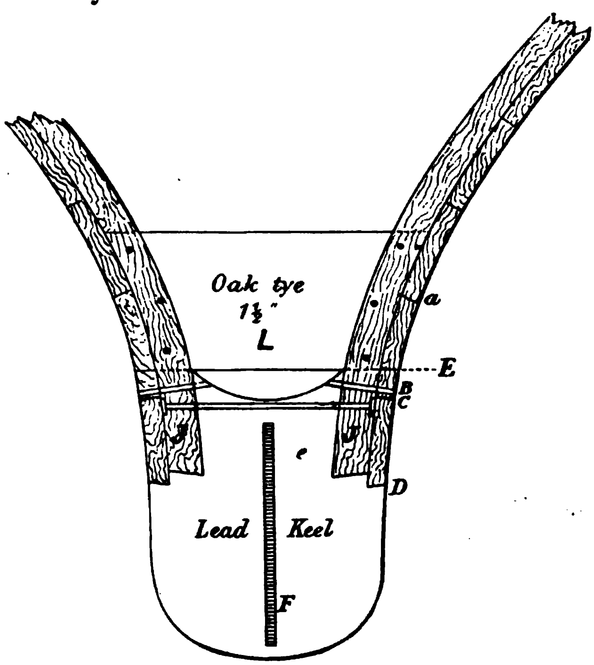


Fig. 182.

In yachts of 70 tons and upwards the heels of the floor should be deep enough to take two through bolts, as shown in Figs. 178 and 179.

Fig. 181 represents the construction of the Saraband 10-tonner at the mid-section, the keel being the broadest put into a 10-tonner.

In the case of small yachts with deep thin lead keels like the Dolphin (Plates XXI. and XXII., Series A), great care has to be taken in boring the lead for the bolts, and in ensuring a true connection with the holes in the wood keel.

Several plans have been proposed for dispensing with the wood keel altogether, but hitherto the plans have not met with favour.

Mr. John Chambers (of Messrs. Page and Chambers, Lowestoft, who

built the Saraband) has contrived one of the most likely plans, by which means the wood keel can be dispensed with, and great weight consequently

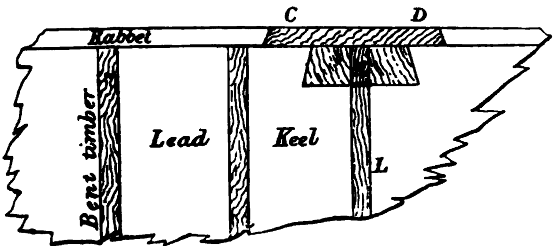


FIG. 183.

saved. In the case of a 10-tonner like Saraband the saving in weight would be about a quarter of a ton. Fig. 182 shows a cross-section view.

a C is the garboard strake.

B, bolt through the lower edge of the garboard, through the lead keel.

E is the line of the top of the lead.

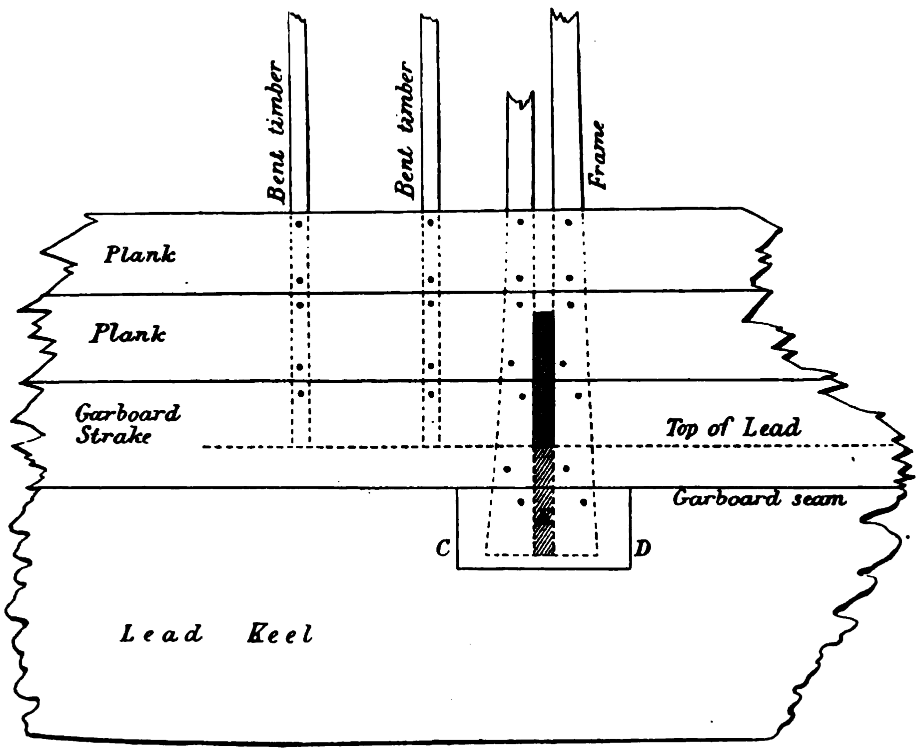


FIG. 184.

F is an iron girder cast into the keel.

J is a frame bolted through the keel, but this seems hardly necessary,

as it will be seen from Figs. 183 and 184, the heels of the frames are so dovetailed that they could no more be forced out than they could be drawn.

C D is a filling piece over heels of frames shown also in Fig. 184.

K is an oak key on the principle of a "Lewis."

L is a piece of 1½in. oak timber which ties the frames together (see Fig. 182).

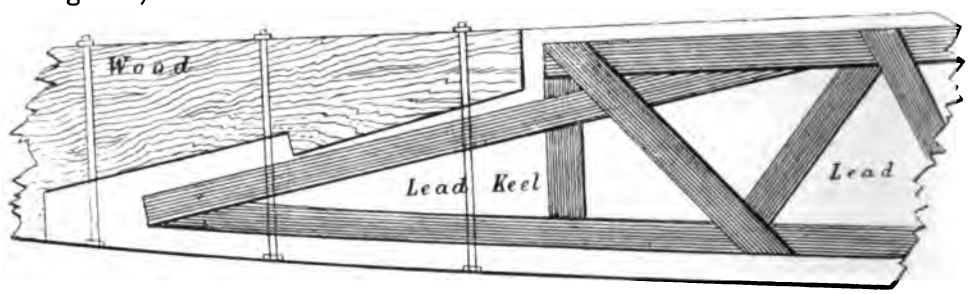


FIG. 185.

Figs. 185 and 186 show the mode of scarphing the wood keel to the lead keel. *d* is a dowell.

If necessary a similar girder could be cast in the keel near the top, so that in sections the girder would show in the ⊤ form. This would provide for lateral stiffness as well as vertical. It should be noted that it would not be safe to trust to the lead alone for stiffness as it bends very readily, and although, in a vessel with sections like Saraband the plank

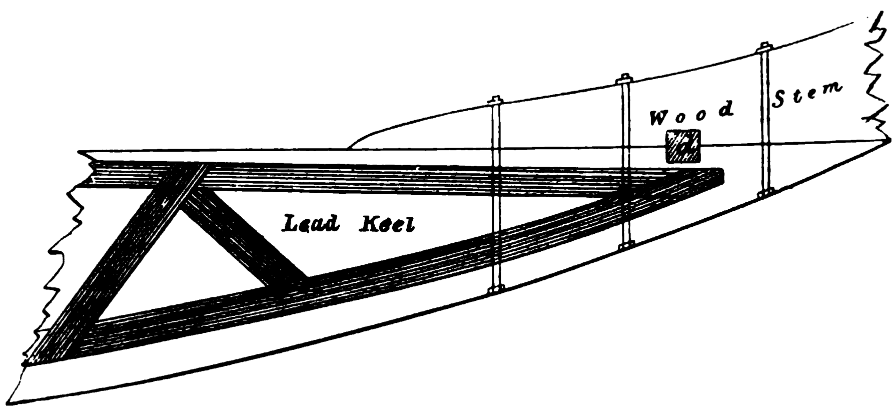


FIG. 186.

greatly adds to the vertical stiffness, the addition of the girders in the lead would be also necessary.

A scarph in a keel is shown by Fig. 187. Sometimes the jog *a* is in the form of a tenon only, but the plan shown is the better, with the tenon *t* in parts of the scarph.

The letter *w* denotes "a stop-water," and is a plug of soft fir driven

tightly into a hole bored through the keel at the upper jog, as shown. When the vessel is put afloat this plug swells and prevents leakage. Stop-waters are also placed in the junction of stem and keel and stern post and keel. The length of scarphs depends greatly upon the depth of the keel, and, as a rule, should make an angle of 13° with the top, or under side. In some of the published rules for building the length of scarphs are given for certain breadth or "siding" of keel; but this obviously is misleading, as the depth or "moulding" of keels vary so.

The "fin bulb" construction is shown on Plates XIII. 2 (A) and XIV. (A), &c.

The keel having been sawn out according to the plans, or from moulds, the next thing to do will be to get it into place on the building blocks.

In laying the blocks to take the keel, care should be taken to ascertain if the ground is quite solid, and the side of the bottom blocks will some-

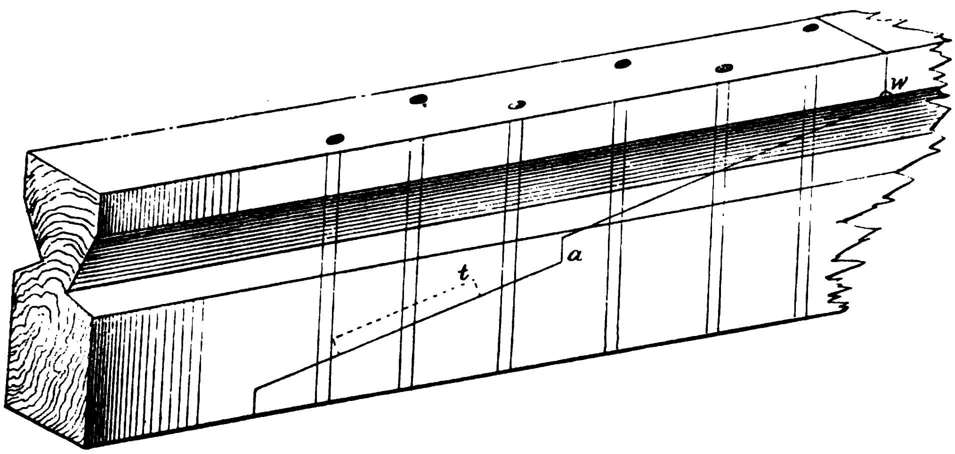


Fig. 187.

what be guided by the condition of the ground. If possible the vessel should be built on a declivity, to simplify the arrangements for launching.

The general practice now is to build with the L.W.L. parallel to the line of the horizon, or just as the vessel will float on the water. The blocks are then so laid that they take the underside of the keel for the required level position. This is easily done by stretching a line at some convenient distance over the blocks to represent the base line, the latter being, as before explained, made parallel to the load water-line by aid of a spirit level.

In fixing on the height of the blocks, due care must be taken to provide facility for getting in the keel fastenings, so as to avoid unnecessary excavations.

When a vessel has a lead keel, it is sometimes the practice to cast this keel and make it form part of the building blocks. The objection to this

is that the floor and keel bolts can then only be put in through the lead keel, and this is objectionable now that lead keels have grown to be

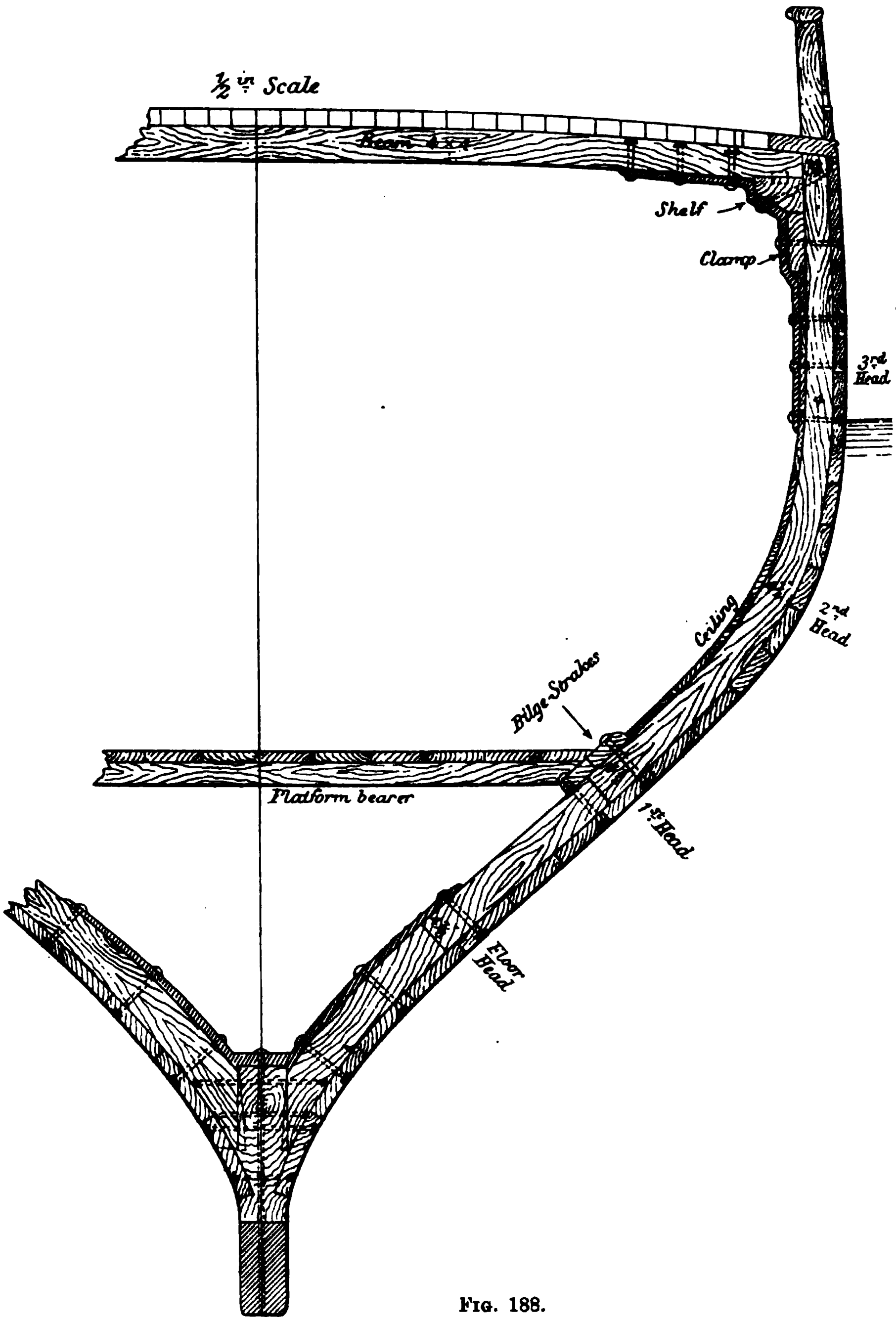


FIG. 188.

so large, mainly on the score of the cost of the bolts and the labour in boring the holes to take them.

In the case of small yachts the stem and stern post are usually fitted to the keel before placing on the blocks by aid of the moulds, and this will be the easier plan as a rule.

A wood frame of a vessel, as will be seen from Plate XIV., XVI., &c., and Figs. 188 and 189, are composed of various parts, and it is usual to so arrange the diagonal lines shown in Fig. 174 that they cross the heads and heels of the timber.

The siding of the frames alter abruptly, as will be seen by Plate XIV., but the moulding tapers gradually as shown in Fig. 188.

The heads and heels of the timbers are usually connected by dowells.

The common practice now is to make the frames close jointed, with square iron bolts for connections, as shown in Plate XIV.; but often the imbers are placed two or more inches apart with a chock between, and

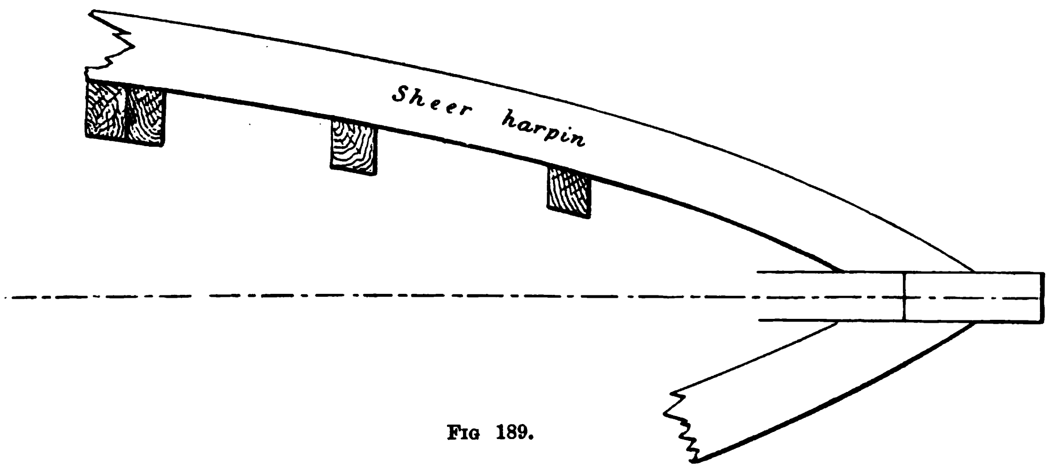


FIG 189.

connected by square galvanised iron. So far as the frames go, those which are close jointed have the greatest strength, but there is some risk of decay owing to the damp getting between the joints; on the other hand the open frames give more support to the plank, especially in small vessels, where there is little depth of seam for caulking.

When the frames are finished and marked they are raised upon the keel and plumbed and squared into position. In a small vessel this is a very easy operation, but not so in a large, and tackles, ropes, and shores will have to be resorted to. When the parts of two frames on opposite sides of the keel are in position, it is a good plan to connect them by a deep batten across the water-line marks, as shown in Fig. 190.

A common practice for facility in getting the frames into their right places in small vessels, is to first fix a sheer harpin by aid of shores and struts from the stem to stern-post. This ribband will represent the deck outline as taken from the half breadth plan. For a 10-tonner, say, it

would be made of inch deal, and be about 10in. wide. The stations for the heads of the timber are marked on it, or cut out as may seem most suitable (see Fig. 189).

Single timbers are usually worked for about one-sixth the deck length in the bow, and, formerly, single timbers were occasionally worked all through. There is no objection to this plan in small vessels of 20 tons and under if the grain runs with the curve of the timbers, and if the timbers are in one length keel to deck. Some large vessels, such as the schooner Alarm, were single framed in varying lengths, great care being taken that there was a proper shift of butts, so that the jointing of heads and heels did not make a continuous straight line in a fore and aft direction. When the midship frame and two or three other frames are in position, a harpin or ribband is put round, as shown in Fig. 190.

These are sometimes square and sometimes triangular in section, the latter bending the truer. These ribbands are secured by coach screws or eye screw bolts to the frames; the latter are to be preferred, as they can be screwed up with greater ease, but they should be used with a flat ring or washer. When the harpins are all on, the framing will be continued, and some fairing with the adze may be necessary. If, however, the timbers were cut out with proper care from these moulds, the chipping will not be required.

In small vessels every other frame is sometimes left out, and a steamed timber worked instead. Particulars of this construction will be found further on.

When the frame is complete the planking will be commenced from the sheer strake downwards, the plank when required being steamed. The strakes are held in position by screw clamps, as shown in Fig. 190, and sometimes chains set up by a sort of tourniquet are resorted to if all the clamps are in use.

Butts should not come nearer than 5 feet to each other on adjoining planks; and no butts are allowed on the same timber unless three strakes come between them. This rule should be very particularly attended to, especially in vessels which have but one or two inside strakes, and but few through fastenings.

With regard to inside strakes, commonly termed bilge stringers, or bilge strakes, there should be a pair worked over the floor heads on each side of vessels under 80 tons, and two pairs each side in vessels above 80 tons, connected by an iron breast-hook at the stem, somewhat of the form of the iron knee floors shown in Fig. 178. The shift of butts in the bilge strakes should not be less than 5ft. In vessels of 80 tons and upwards a wale is usually worked, or thick strake outside just above the

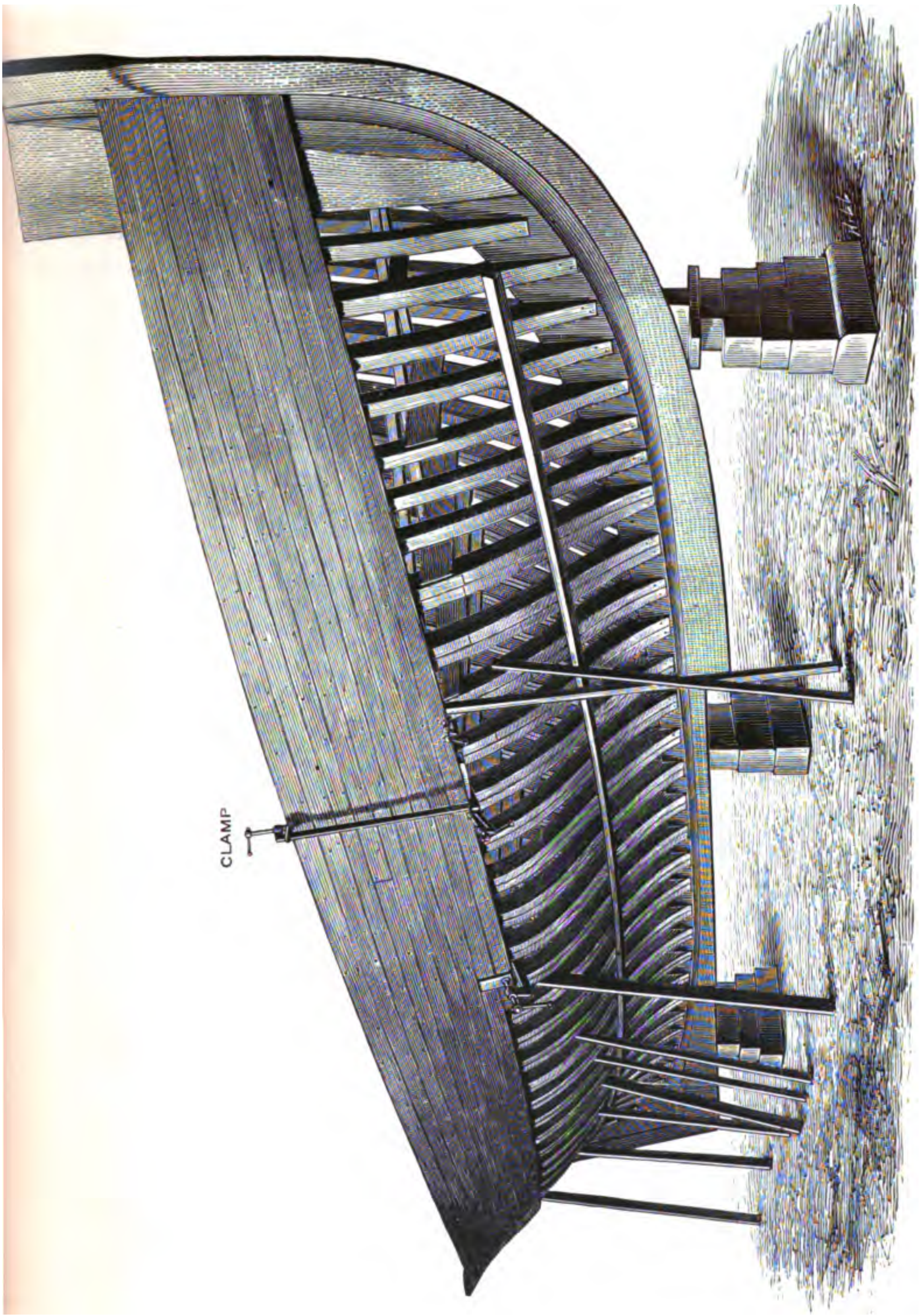


FIG. 190.

water-line. The size for deck beams will be taken from the table, but the full size should not run from stem to stern, but get smaller as the beam decreases at either end, except at the mast and windlass.

The arch given to beam varies, being most in ~~small vessels~~; generally ¼in. to the foot is allowed.

It is seldom that a shelf is met with in 5-tonners, a clamp only doing duty as a shelf to take the deck-beams and fastenings and timber-head fastenings. When a clamp only is fitted in a 5-tonner, the top strake (under the covering board) should be of oak or teak, and bolted through timber, clamp and all. See Vivid, Plate XIV. 2.

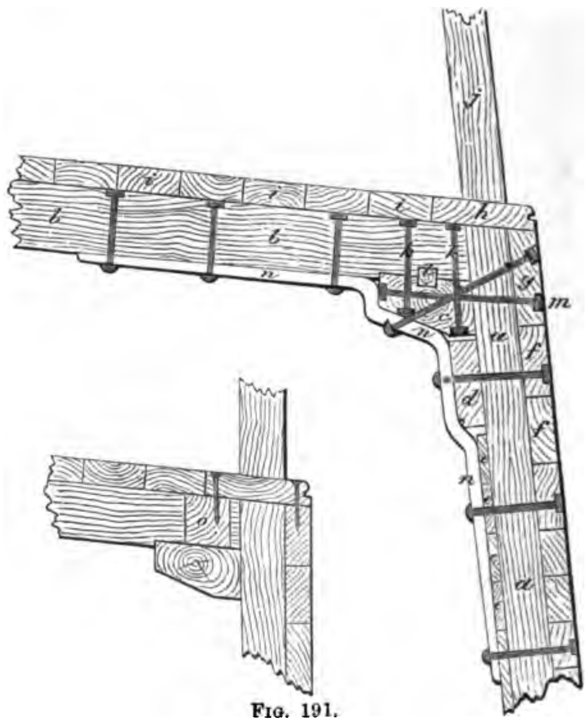


Fig. 191.

The correct method of construction for this part of the vessel is shown by Fig. 191; *a* is a timber of the frame; *b* a deck beam; *c* the shelf; *d* the clamp; *e* the ceiling; *f* the outside plank; *g* the top strake or sheer strake; *h* the covering board; *i* the deck plank; *j* a stanchion fitted between the frames; *k k* bolts through every beam; *l* a dowell; *m* a bolt through every frame; *n* a hanging knee bolted as shown; *o*, in the other sketch, shows a section of pieces of timber fitted between deck beams to take the fastenings of the covering board. Each piece of timber is bolted to the shelf.

The number of hanging knees depends upon the size of the yacht, and the minimum number allowed is set forth in the table farther on.

The shelf will be secured to the knighthead and stem forward by a galvanised iron or oak breasthook of about the same size as the hanging knees. In the wake of the masts, and at intervals of fifths of the length, lodging knees of oak will be worked under the deck to the beams.

The construction of the counter is shown by Plate XIV. 3.

A A A, &c., are chocks to take the ends of the deck (see P, Fig. 192).

B B, rudder trunk boards rabbetted into horn timbers, and extending to under deck. The boards are secured to the stern post by the bolts *d d d*, &c. It will be seen that there is a filling piece between the stern post and trunk boards, to insure the rudder post not jamming, and also to leave room to work a caulking iron if necessary. For this purpose it is usual to leave a good space abaft the rudder post.

D is the thwartship chock which forms the aft side of the rudder trunk, as shown by L in Fig. 192.

C is a long chock, to take the ends of the plank (see *e*, Fig. 193); *f* is the stern post, and *h* the inner post. The rake of the post is allowed

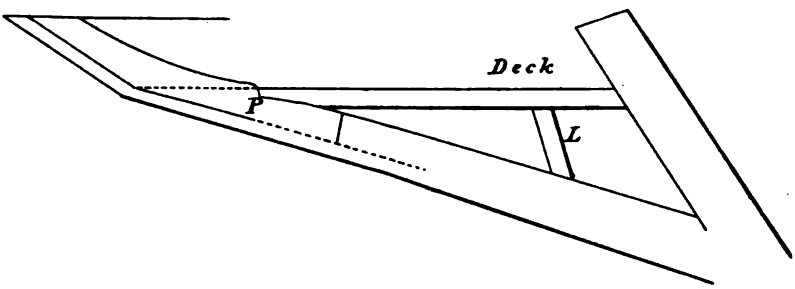


FIG. 192.

for, as will be seen; *s s s*, &c., are the fastenings of shelf through frames and plank.

X Stanchions.

The manner of working the quarter timbers are shown on Plate XIV. 3. Another plan is shown by Fig. 194.

The quarter timbers are tenoned into the head of the transom frame (see Fig. 194).

The horn timbers are bolted through the stern post, and tenoned into the frame next ahead of the transom frame at J. They are shown also in section K in Fig. 194.

In Fig. 194 (near the end of the counter) the manner of jogging the cross pieces, S, is shown by E D, also the probable section of the quarter timbers at that part (see N, Fig. 194).

The part of the quarter timber above A (see R in the sheer outline, Fig. 194) is usually fashioned by a moulding termed a quarter badge, but a more elegant finish is to work it off flush with the plank.

The metal fastenings of yachts (see table farther on) form a subject about which there is a great deal of opposite opinion. The builder who constructs cheaply contends stoutly that there is nothing like iron for above-water fastening; on the other hand, the builder who always asks and gets a good price for his work would as soon think of fastening a yacht with cabbage

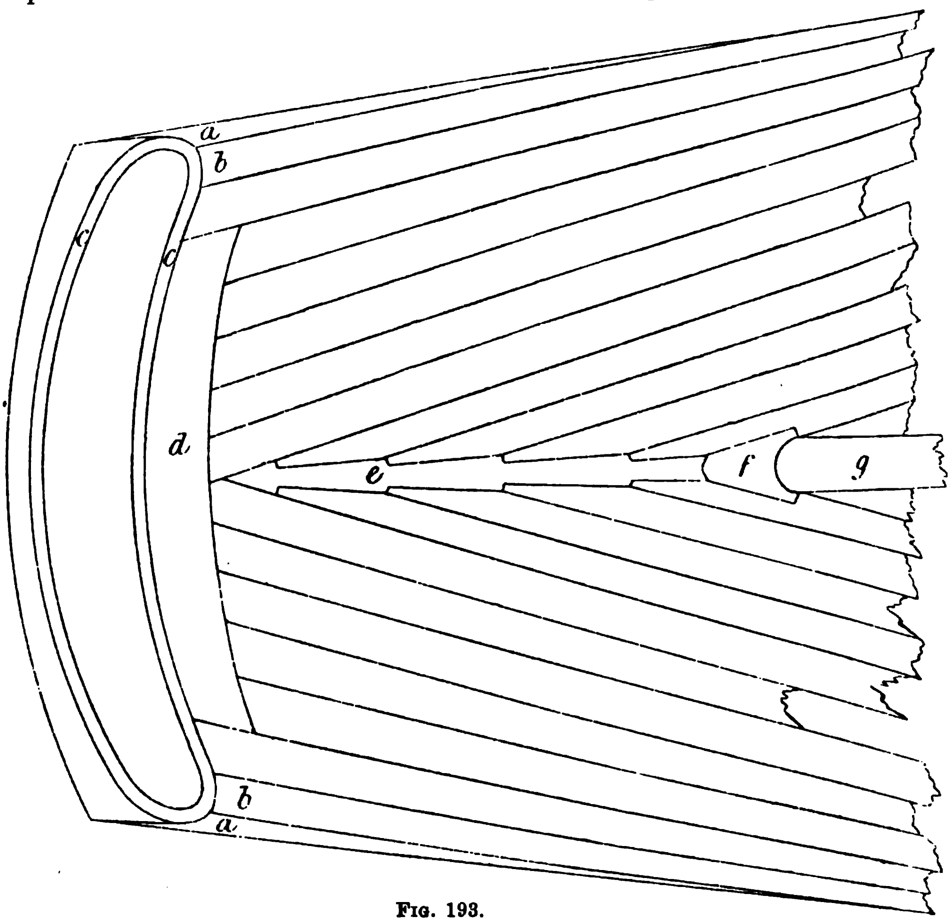


FIG. 193.

¼in. Scale.

METHOD OF BUTTING AFTER ENDS OF PLANK.

a Flare off of bulwark.
b Top strake.
c Stern moulding.
d Plank worked athwartships.
e Chock rabbeted to take ends of plank.
f Stern port rabbeted into plank and stern chock and secured with screws.
g Rudder.

stumps as use any iron bolts at all. Others recommend iron for dead-wood bolts, shelf-bolts, and floor bolts, with Muntz metal for plank fastening. The most approved plan is to have Muntz metal of good quality for all long bolts, and copper bolts for all scarphs and plank butt fastenings. There is no doubt that iron has advantages, so far at least as long dead-wood bolts

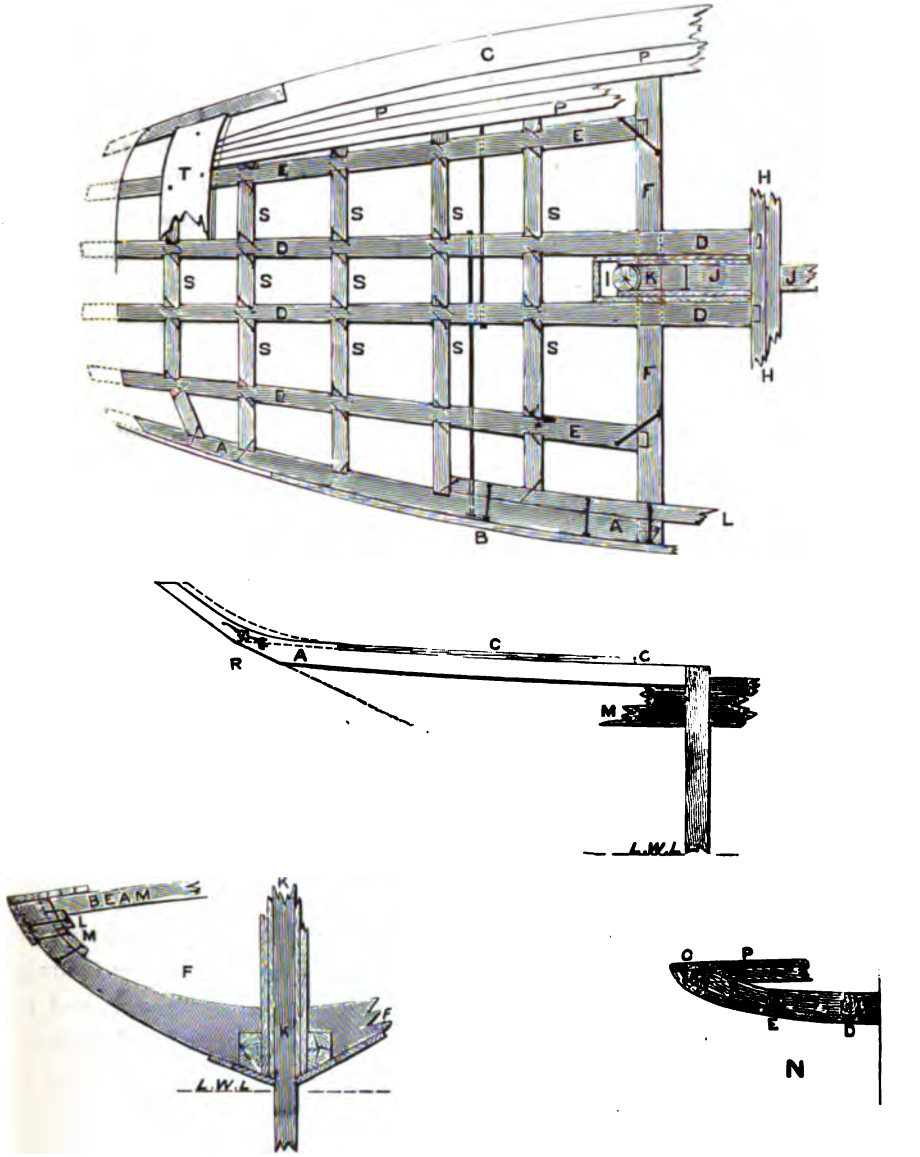


FIG. 194.

A Quarter timber.
B Top Strake.
C Covering Board.
D Horn timbers bolted through sternpost.
E Stern timbers.
F Transom Frame.
H Frame.
I Rudder trunk.
J Deadwood.
K Sternpost, not allowing for rake, showing section of horn timbers each side.
L Shelf.
M Clamp.
N Sketch, showing jogging of cross pieces.
P Deck.
S Cross pieces.
T Chock to receive ends of deck.

are concerned. Iron can be driven very much tighter than copper rod, and its strength is greater. We have seen iron bolts, that had been made hot and dipped in varnish or oil before they were driven, taken out of a vessel thirty years old, long before galvanised iron was heard of, as clean and bright—in fact, the varnish unperished—as they were at the moment of driving. On the other hand, if the iron were driven through a shaky piece of timber, or loosely driven, so that salt water might get to it with drainings from the copper, decay of the iron would be very rapid ; hence preference must be given to copper or Muntz metal, but the latter if of inferior quality is very brittle. Galvanised wire is often used for shelf fastenings, and it is less objectionable there; but preference should be given to Muntz metal for all dead woods, as it can be driven as tightly as iron, and, if of the best quality, will clench as well. Copper for scarphs of keel, and all butt bolts.

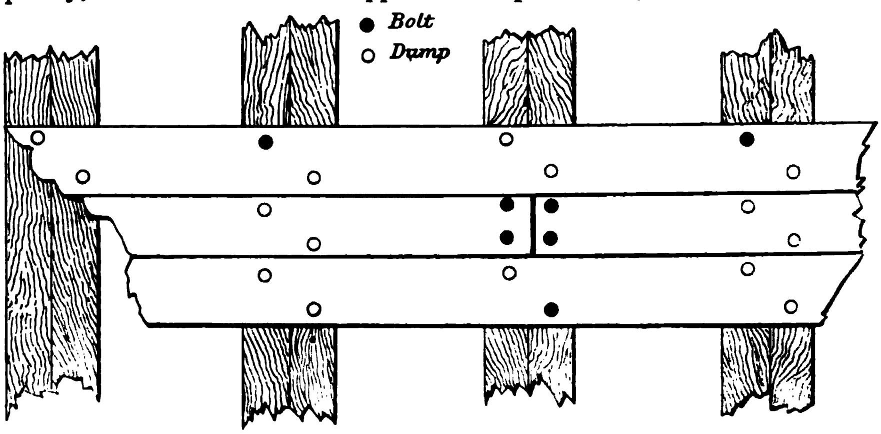


FIG. 195.

Whether trenails or bolts are used, one should be put in each plank or strake at every timber; two-thirds of the trenails must be driven through, and the butts of the plank should be through-bolted with copper; and the bilge strakes and limber strake should also be through-bolted with copper.

Trenails, owing to the large holes they require, are not adapted for vessels of less than 50 tons on account of the small size of the timbers, and generally should only be used in the bottom plank.

The system of fastening with one through bolt to every other frame and a dump to the remaining timbers is shown by Fig. 195.

In some cases only one dump is used for each frame instead of to each timber, and only one through bolt for the butts ; or one through bolt and a dump. Lloyd's require the plank to be double fastened (as shown in Fig. 195) if the plank be 11in. broad ; double and single if 8in., and under 8in. single. The dumps in length should be at least double the thickness of the plank.

Care must be taken that the seam of the butt comes on the face of a timber and not on the joint, so that there may be a stop for the caulking.

The shift of plank for the deck will be the same as for the outside plank. The deck planks are dowelled together and fastened diagonally from one side only into the beam. The dowells are worked horizontally, and in some well-built vessels a tongue of hard wood is inserted in the butt ends. Galvanised iron or yellow metal dowels and nails are used. If ungalvanised iron is used rust will soon show through the deck.

In the case of steam yachts, round the steel or iron coamings a strake of teak should be worked, otherwise the rust from such coamings will stain the fir plank, and such stains are difficult to remove.

It will be seen from Plate XIV. that the deck plank is made to taper; and forward the ends are now usually square butted into the covering board, and not finished with a feather edge as shown in the Plate. The main object of thus butting the ends of the plank into the covering board is to avoid a thick ugly caulking seam, and perhaps a leaky one, round the covering board. The butts are usually about one-third the breadth of the plank.

In caulking very great care should be taken to drive the oakum equally as to hardness. For this reason too many hands should not be employed in caulking, especially on the same seam, as no two men will drive it alike. The oakum should be driven as far through the seams as practicable.

Cotton is usually used for deck seams, and great care should be exercised in driving it so that the edges of the plank are not injured.

In the table of years assigned to different material it should be noted that American elm, or English elm, is liable to very rapid decay if subjected to alternate wetting and drying, or to ill ventilation or damp air.

In giving a class and term of years to a yacht by Lloyd's it is generally done under what is known as the "mixed material" rule. That is to say the timber of lowest grade used is taken as the basis, and then for the superior timber used and good workmanship one or two years are added. An extra year is also given if a yacht has no iron fastenings below the load-line, but a mixture of treenails and yellow metal or copper; and if iron is excluded entirely from the outside two extra years are given; if the whole of the fastenings are of yellow metal and copper, three extra years are given. An extra year is also given if the yacht is built under a roof. If, then, a yacht has stem, stern post, frames, and dead wood of English oak, and plank of pitch pine below the load-line, and teak above,

her grade would be twelve years. If all fastenings were of yellow metal or copper, her term of years would be made up as follows:

Term for material used	12 years
Mixed material rule........................	1 year
Material fastenings	3 years
Built under roof	1 year
	17 years

If the bottom planking is of English or American elm, and the shelf of teak, another year would be given under the mixed material rule. Sometimes the additional year is given where the shelf is not of teak, providing an equivalent quantity of teak is worked in elsewhere—such as for bilge strakes or clamps.

The letter A denotes the character and condition of the materials used as being good. The numeral 1 denotes that the equipment of the yacht, such as her gear, blocks, anchors, chains, hawsers, &c., are good, and of the sizes required by the rules. Lloyd's, however, are only particular as to the size and strength of blocks and ropes, and pay no attention to their finish or appearance.

NUMBER OF YEARS ASSIGNED BY LLOYD'S TO DIFFERENT KINDS OF TIMBER.

Description of Timber.	Keel.	Stem, Sternpost, Apron, Inner Sternpost, Deadwood, Knightheads, Hawse Timbers, Frame Timbers, Beams, and Hooks.	Outside Planking.		Shelves, Clamps, Limber, and Bilge Strakes, and Keelsons.	Upper Deck, Waterway, Covering-board, and Boughtree Timbers.	Rudder, Windlass, and Pulbitt.—Mainpieces.
			From top of Keel to two feet below Load-line.	From two feet below Load-line to Planksheer.			
East India Teak	16	16	16	16	16	16	16
English, African, French, Adriatic, Italian, Spanish, and Portuguese Oaks, Greenheart, Morra, and Iron Bark.........	12	12	12	12	12	12	12
Pitch Pine, Oregon and Huon Pine, Larch, Hackmatack, and Cowdie or Kaurie Pine	—	—	12	10	10	10	—
Northern Continental Oak......	10	9	12	10	10	10	10
Dantzic, Memel, Riga, and American Red Pine...........	-	—	9	9	9	10	—
English Elm and American Rock Elm	16	—	16	—	—	—	—
Spruce Fir, Swedish and Norway Red Pine	—	—	8	8	—	—	-

The tables which follow are those which were compiled for the Yachting Lloyd's in 1876, and are as suitable as those given in Lloyd's Register; but if a yacht is to be classed, Lloyd's rules would have to be complied with. [Which Rules can be obtained from Lloyd's, White Lion-court, Cornhill.]

SIZES OF TIMBERS, KEELSON, KEEL, PLANKING, &c. (NOT LLOYD'S.)

Tonnage, Y.B.A.	3	5	10	20	30	40	50	75	100	150	200	250	300
	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.
Timber and Space (double timber frames)	15	16	17	18	19	20	21	22	23	24	25	26	27
Floors, sided * at keelson	2¼	2½	3	3½	4	4½	5	5½	6	7	7½	8	8½
Floors moulded at keel	2¾	3	3½	4	4½	5	5¼	6	6½	7¼	7¾	8½	9
Futtocks (timbers), sided and moulded at floorheads	2	2	3	3	4	4	4½	5	5½	6½	7	7	8
2nd Futtocks, sided	—	—	—	3¼	3¾	4¼	4¾	5	5½	6	6½	7	7½
Top Timbers sided	1¾	2	2¼	2¾	3	3½	3¾	4¼	4½	4¾	5	5¼	5½
Top Timbers, moulded at heads	1½	1¾	2	2¼	2¾	3	3¼	3½	3¾	4	4¼	4½	4¾
Keel, Stem, Apron, Sternpost, and Deadwood, sided and moulded	4	4	4½	5½	6	7	7½	8½	9	9½	10	10½	11
Keelson, sided and moulded	4	4½	5	6	7	8	9	9½	10	11	11½	12	12½
Wales, ⅙ depth of ship	—	—	—	2¼	2½	2⅝	2¾	2¾	3	3½	3¾	3¾	4
Bottom Plank, from keel to wales	1	1	1¼	1½	1¾	1⅞	2	2	2¼	2¼	2½	2½	2½
Clamp	1½	1½	1¾	2	2	2⅛	2¼	2¼	2½	3	3¼	3½	3½
Ceiling	½	½	⅝	¾	⅞	1	1⅛	1¼	1¼	1½	1¾	2	2¼
Bilge Plank or Stringer inside, over floor heads	1¼	1¼	1½	1¾	2	2¼	2½	2½	2½	3	3¼	3¼	3½
Shelf, sided and moulded, or of equivalent sectional area	2½	3	3½	4	4¾	5	5¾	6	6¼	6¾	7¼	7½	7¾
Flat of Deck	1¼	1¼	1½	1¾	2	2	2¼	2¼	2½	2½	2¾	2¾	2¾
Scarph of Keel or Keelson (see page 453)	5ft.	5ft.	5ft 3in	5ft 6in	5ft 9in	6ft	6ft	6ft 3in	6ft 6in	6ft 9in	7ft	7ft	7ft

* Sided means the thickness of the timber between its two straight or flat sides; and "moulded" the thickness between the curved sides on which the planking is fastened. Sometimes, if the siding is increased a little beyond that given in the table, the moulding is proportionately decreased.

The butts of planks to have not less than five feet shift, unless two strakes of plank intervene between the two when the butts occur on the same frame or timber.

H H

SIDING AND MOULDING OF BEAMS.

Length of Beam. Amidships.	Lower Deck Beams. Sided and Moulded.		Deck Beams Moulded. Sided Amidships.		Moulded at Ends.	Spacing.
ft.	in.	in.	in.	in.	in.	in.
5	—	—	2 1/2	2 1/4	2	21
6	—	—	3	2 3/4	2 1/2	22
7	—	—	3 1/4	3 1/4	3	23
8	—	—	3 3/4	3 1/2	3 1/4	24
9	—	—	4	3 3/4	3 1/2	25
10	—	—	4 1/2	4	3 3/4	26
11	—	—	5	4	3 3/4	28
12	—	—	5 1/4	4 1/4	3 3/4	30
13	5	5 1/2	5 1/2	4 1/2	4	32
14	5 1/2	6	5 3/4	4 3/4	4	34
15	6	6 1/2	6 1/4	5 1/4	4 1/2	36
16	6 1/2	7	6 1/2	5 1/2	4 3/4	38
17	6 1/2	7	6 3/4	5 3/4	5	39
18	6 3/4	7 1/4	7	6	5 1/4	40
19	7	7 1/2	7 1/4	6 1/4	5 1/2	41
20	7 1/4	7 3/4	7 1/2	6 1/2	5 3/4	42
21	7 1/2	8	7 3/4	6 3/4	6	43
22	7 3/4	8 1/4	8	7	6	44
23	8	8 1/2	8 1/4	7 1/4	6 1/4	45
24	8 1/4	8 3/4	8 1/2	7 1/2	6 1/2	46
25	8 1/2	9	8 1/2	7 3/4	6 3/4	47
26	8 3/4	9 1/4	8 3/4	8	7	48
27	9	9 1/2	8 3/4	8 1/4	7 1/4	48
28	9 1/4	9 3/4	9	8 1/2	7 1/4	48
29	9 1/2	10	9 1/4	8 3/4	7 1/2	48
30	9 3/4	10 1/4	9 1/2	9	7 3/4	48

METAL FASTENINGS.

Tonnage, Y.R.A.	3	5	10	20	30	40	50	75	100	150	200	250	300
	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.
Keel, knee, stemson, and deadwood bolts.	6/16	6/16	7/16	8/16	9/16	10/16	11/16	14/16	15/16	1	1	1 1/16	1 3/16
Bolts, scarphs of keel, arms of breast-hooks, riders, hanging knees to deck beams, and in and out bolts of shelf and clamp	3/8	3/8	3/8	6/16	8/16	9/16	11/16	12/16	12/16	12/16	12/16	12/16	14/16
Keelson bolts (on through keel at each floor), throats of transoms, of breast-hooks, and of hanging knees	6/16	6/16	7/16	8/16	10/16	11/16	12/16	12/16	13/16	14/16	15/16	15/16	1
Bilge, limberstrake, and through butt bolts	3/8	3/8	3/8	7/16	8/16	9/16	9/16	9/16	10/16	10/16	11/16	11/16	12/16
Other butt bolts	3/8	3/8	3/8	6/16	7/16	8/16	9/16	9/16	10/16	10/16	10/16	11/16	11/16
Bolts through heels of timbers abutting on sides of dead wood fore and aft	3/8	3/8	3/8	6/16	8/16	9/16	10/16	10/16	11/16	11/16	11/16	12/16	12/16
Pintles of rudder	1 1/8	1 1/8	1 1/4	1 1/2	1 5/8	1 3/4	1 7/8	1 7/8	2	2	2 1/4	2 3/8	2 1/2
Rudder braces	5/16	5/16	6/16	6/16	7/16	7/16	8/16	8/16	9/16	10/16	10/16	11/16	11/16
	1 1/2	1 1/2	1 3/4	2	2 1/4	2 1/4	2 1/2	2 3/4	3	3 1/2	4	4 1/4	4 1/2
Hardwood trenails	—	—	—	—	—	—	1	1	1	1	1 1/8	1 1/8	1 1/8
Lead keel centre bolts	1 1/4	1 1/4	1 3/8	1 1/4	1 3/8	1 1/2	1 5/8	1 3/4	2	2	2 1/8	2 1/4	2 3/8
Spaced (in inches)	30	32	34	18	19	20	21	22	23	24	25	26	27
Lead keel side or wing bolts	7/8	7/8	7/8	7/8	1 1/8	1 1/4	1 1/4	1 3/8	1 1/2	1 1/2	1 1/2	1 1/2	1 1/2
Spaced (in inches)	30	32	34	18	19	20	21	22	23	24	25	26	27
Weight of lead keel in tons	5	7	11	18	25	32	40	60	80	100	100	100	100

If the lead keel bolts come in the way of a floor knee bolt they will be then put in diagonally in a fore and aft direction.

NUMBER OF HANGING KNEES.

Tons.	Hold Beams.	Upper Deck Beams.	Tons.	Hold Beams.	Upper Deck Beams.
	Pairs.	Pairs.		Pairs.	Pairs.
5	—	3	75	—	6
10	—	3	100	3	7
20	—	4	150	4	9
30	—	4	200	6	11
40		5	250	8	13
50		5	300	10	15

ANCHORS FOR SAILING YACHTS.

Thames Tonnage.	Number of Anchors	1st Anchor.		2nd Anchor.		3rd Anchor.		4th Anchor.	
		Weight.	Test.	Weight.	Test.	Weight.	Test.	Weight ex Stock.	Test.
			Tons.		Tons.		Tons.	Cwts.	Tons.
3	2	With Stock 75lb.	—	With Stock 28lb.	—	—	—	—	—
		Cwts.							
5	2	With Stock $\frac{3}{4}$	—	With Stock 40lb.	—	—	—	—	—
				Cwts.					
10	2	With Stock 1	—	With Stock $\frac{1}{2}$	—	With Stock 20lb.	—	—	—
20	2	With Stock $1\frac{1}{4}$	—	With Stock $\frac{3}{4}$	—	With Stock 35lb.	—	—	—
30	2	Ex Stock $1\frac{1}{2}$	$3\frac{13}{20}$	With Stock 1	—	With Stock 50lb.	—	—	—
40	2	2	$4\frac{10}{20}$	With Stock $1\frac{1}{4}$	—	With Stock $\frac{1}{2}$cwt.	—	—	—
50	3	$2\frac{1}{2}$	5	Ex Stock $2\frac{1}{4}$	$4\frac{15}{20}$	With Stock $\frac{3}{4}$	—	—	—
60	3	3	$5\frac{10}{20}$	$2\frac{3}{4}$	$5\frac{5}{20}$	With Stock 1	—	—	—
75	3	$3\frac{3}{4}$	$6\frac{3}{20}$	$3\frac{1}{2}$	$5\frac{18}{20}$	With Stock $1\frac{1}{4}$	—	—	—
100	3	$4\frac{1}{2}$	$6\frac{17}{20}$	$4\frac{1}{4}$	$6\frac{13}{20}$	With Stock $1\frac{1}{2}$	—	—	—
125	3	5	$7\frac{7}{20}$	$4\frac{3}{4}$	$7\frac{3}{20}$	Ex Stock $1\frac{1}{2}$	$4\frac{4}{20}$	—	—
150	4	$5\frac{1}{2}$	$7\frac{16}{20}$	$5\frac{1}{4}$	$7\frac{11}{20}$	2	5	1	—
200	4	$6\frac{1}{4}$	$8\frac{10}{20}$	6	$8\frac{5}{20}$	$2\frac{1}{2}$	$5\frac{5}{20}$	1	—
250	4	7	$9\frac{5}{20}$	$6\frac{1}{2}$	$8\frac{15}{20}$	3	$5\frac{10}{20}$	$1\frac{1}{4}$	—
300	4	8	$10\frac{3}{20}$	$7\frac{1}{2}$	$9\frac{13}{20}$	$3\frac{1}{2}$	$5\frac{18}{20}$	2	$4\frac{10}{20}$

CHAINS FOR SAILING YACHTS.

Thames Tonnage.	Chain Cable.				Stream Chain.		Hawser.		Warp.	
	Minimum Size.	Test.	Breaking Test.	Length.	Length.	Size.	Length.	Size.	Length.	Size.
	Ins.	Tons.	Tons.	Fathoms.	Fathoms.	Ins.	Fathoms.	Ins.	Fathoms.	Ins.
3	$\frac{5}{16}$	—	—	50	—	—	40	3	40	2
5	$\frac{6}{16}$	—	—	50	—	—	40	$3\frac{1}{2}$	40	$2\frac{1}{2}$
10	$\frac{7}{16}$	$3\frac{5}{10}$	$5\frac{1}{4}$	60	—	—	45	4	45	3
20	$\frac{8}{16}$	$4\frac{5}{10}$	$6\frac{3}{4}$	60	—	—	50	$4\frac{1}{2}$	45	3
30	$\frac{9}{16}$	$5\frac{8}{10}$	$8\frac{1}{4}$	60	—	—	60	$4\frac{1}{2}$	45	3
40	$\frac{10}{16}$	7	$10\frac{1}{2}$	75	—	—	60	5	50	S
50	$\frac{10}{16}$	7	$10\frac{1}{2}$	90	—	—	75	5	60	$3\frac{1}{2}$
60	$\frac{11}{16}$	$8\frac{5}{10}$	$12\frac{3}{4}$	105	—	—	75	$5\frac{1}{2}$	75	$3\frac{1}{2}$
75	$\frac{11}{16}$	$8\frac{5}{10}$	$12\frac{3}{4}$	120	—	—	75	$5\frac{1}{2}$	75	$3\frac{1}{2}$
100	$\frac{12}{16}$	$10\frac{1}{8}$	$15\frac{1}{8}$	135	—	—	75	6	90	$3\frac{1}{2}$
125	$\frac{12}{16}$	$10\frac{1}{8}$	$15\frac{1}{8}$	150	45	$\frac{8}{16}$	75	6	90	4
150	$\frac{13}{16}$	$11\frac{7}{8}$	$17\frac{8}{10}$	150	45	$\frac{8}{16}$	75	6	90	4
200	$\frac{14}{16}$	$13\frac{3}{4}$	$20\frac{5}{8}$	150	45	$\frac{9}{16}$	75	$6\frac{1}{2}$	90	4
250	$\frac{15}{16}$	$15\frac{8}{10}$	$23\frac{7}{10}$	165	45	$\frac{9}{16}$	75	$6\frac{1}{2}$	90	$4\frac{1}{2}$
300	1	18	27	165	45	$\frac{9}{16}$	75	7	90	$4\frac{1}{2}$

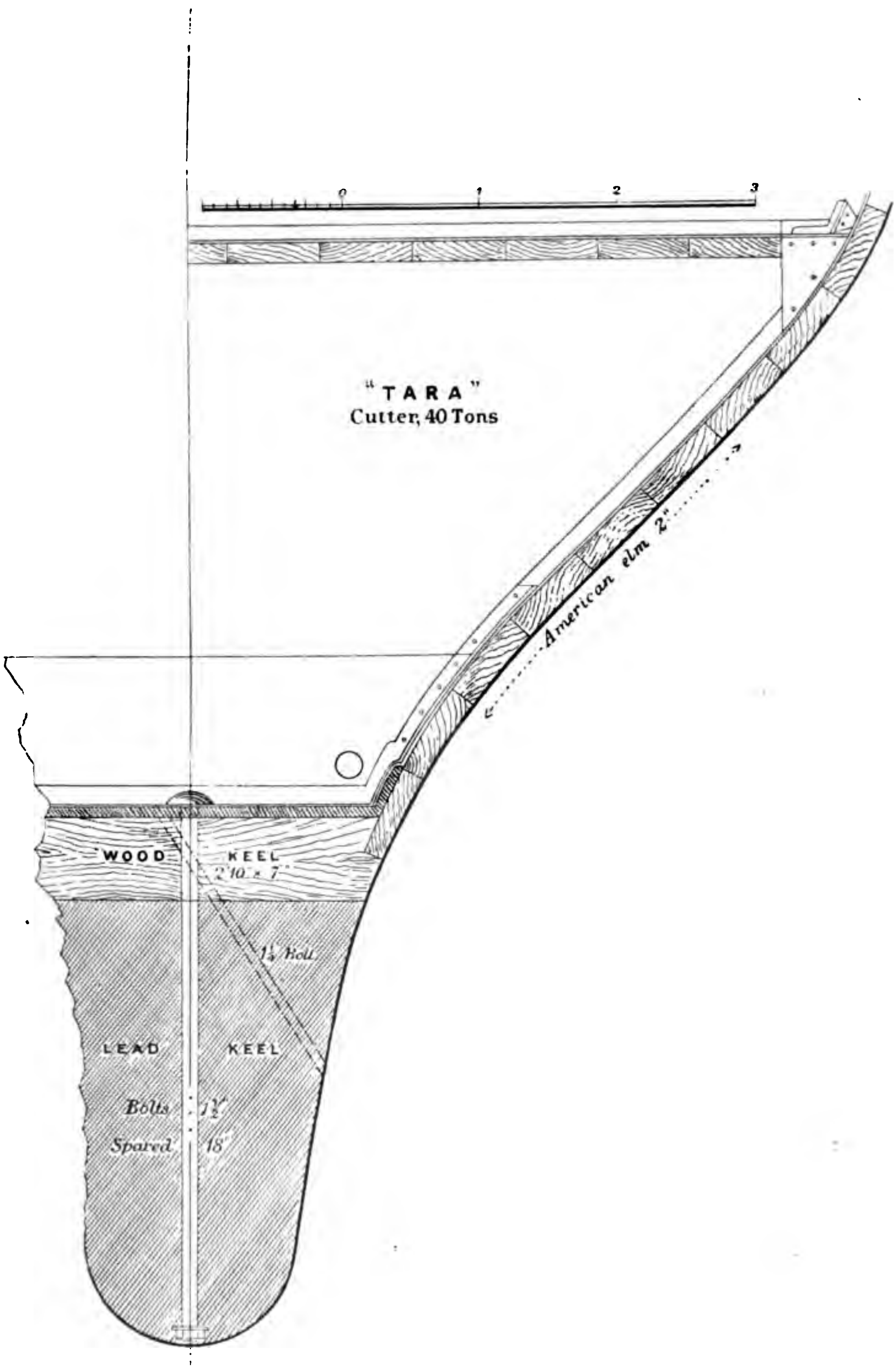
The weight of chain cable per fathom can be found by multiplying the square of the diameter (diameter^2) by 54.

COMPOSITE CONSTRUCTION.

A yacht of composite construction is one which has stem, sternpost, keel, and planking of wood, and frames, beams, floors, gunwales, &c., of iron or steel. The great advantage of this form of construction is that for any given tonnage so much more internal space is available for cabin accommodation in consequence of the smallness of the frames compared with wooden frames. This difference is mostly apparent in large yachts, however, although a considerable advantage exists for small yachts. For a wood yacht of 50 tons the frames would be 4½in. moulded, whereas the moulding depth of the angle steel frame would be only 2in.; for a wood yacht of 300 tons the frames would be 7½in. moulded, whilst the moulding depth of the angle steel frame would be only 3in. Thus in a yacht of 300 tons a gain of at least 6in. breadth could be obtained by adopting the composite system of construction.

It should be noted that the size of the frames, &c., for composite yachts are not determined by tonnage, but by the sum of the depth, girth, and breadth, as for yachts built wholly of iron or steel. In composite yachts, however, the frames must not be spaced more than 18 inches apart, no matter what the size of the yacht is. This practice is followed on account of the caulking, there being nothing to caulk against like the back of the wood frames; and, moreover, the plank not having the support of the broader wood frames, and the fastenings being necessarily farther apart longitudinally, there would be danger of the yacht working the caulking out if the spacing were increased. However, if the yacht has two thicknesses of plank (which is rarely the case) the spacing can be increased.

To further ensure that the composite yacht shall not work, Lloyd's require her to be constructed pretty much the same as an iron or steel yacht. That is to say, she must have deck stringers, tie plates, sheer strakes, diagonal plates, bilge plates, &c., and when these are on she might almost as well have been entirely plated. In many instances of building in which the author has been engaged, Lloyd's have allowed the diagonal plates to be dispensed with by the addition of another outside fore-and-aft plate (see Plates XX., &c.). There is not much doubt that the outside longitudinal plates (as they cross the frames at right angles and are much deeper) are to be preferred to the diagonal tie plates, and, on the whole, they are not so much trouble to plank over as the diagonal plates. Examples of the floor and gunwale construction and longitudinal strengthening by diagonal plates are given in Figs. 196, 197, and 198, representing the construction of the Tara and Milly. Plate XXI. shows a similarly



0
1
2
3
"TARA"
Cutter, 40 Tons
American elm 2"
WOOD
KEEL
2'10" x 7"
1¼" Bolt
LEAD
KEEL
Bolts 1½"
Spaced 18"

Fig. 196.

"TARA," 40-TONS.

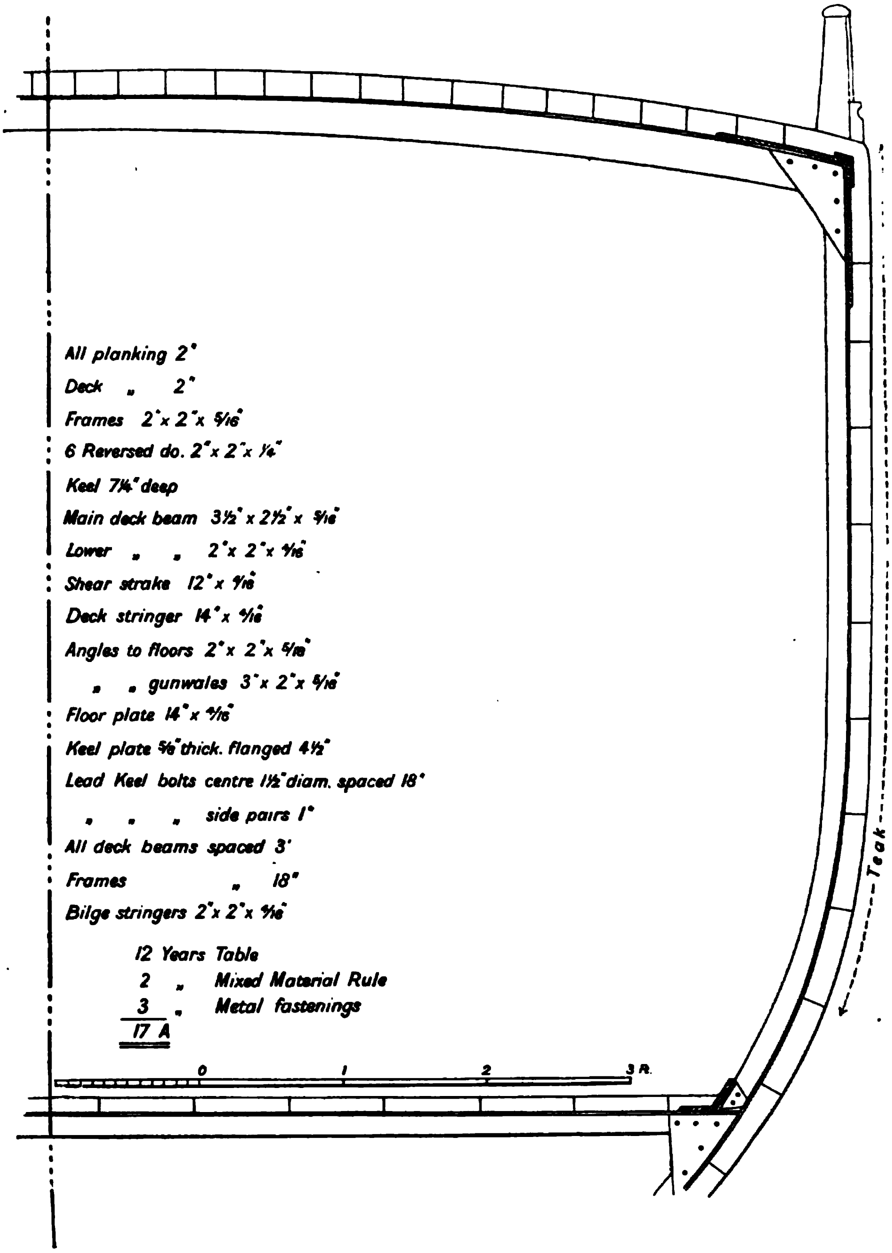


FIG. 197.

"TARA," CUTTER, 40-TONS.

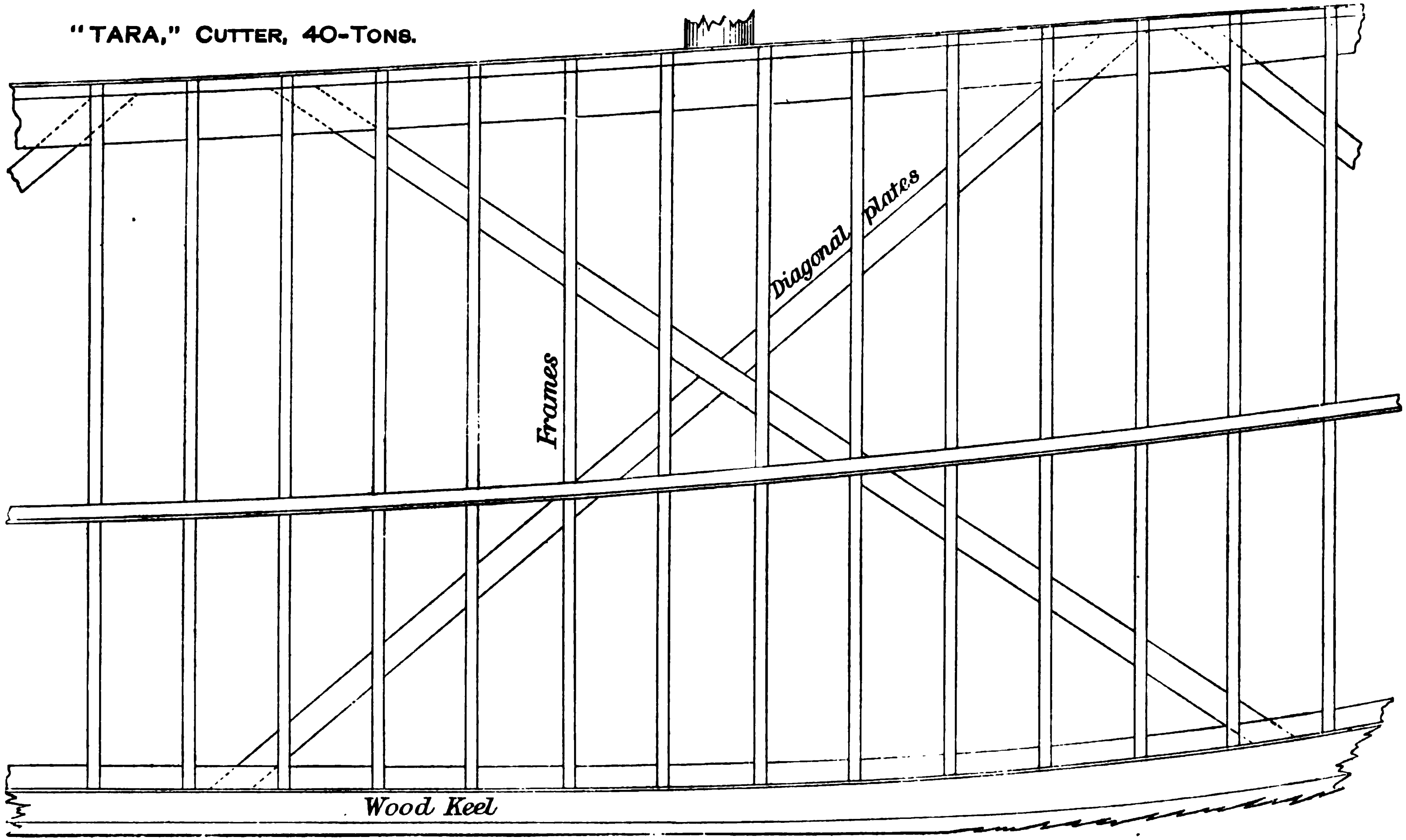
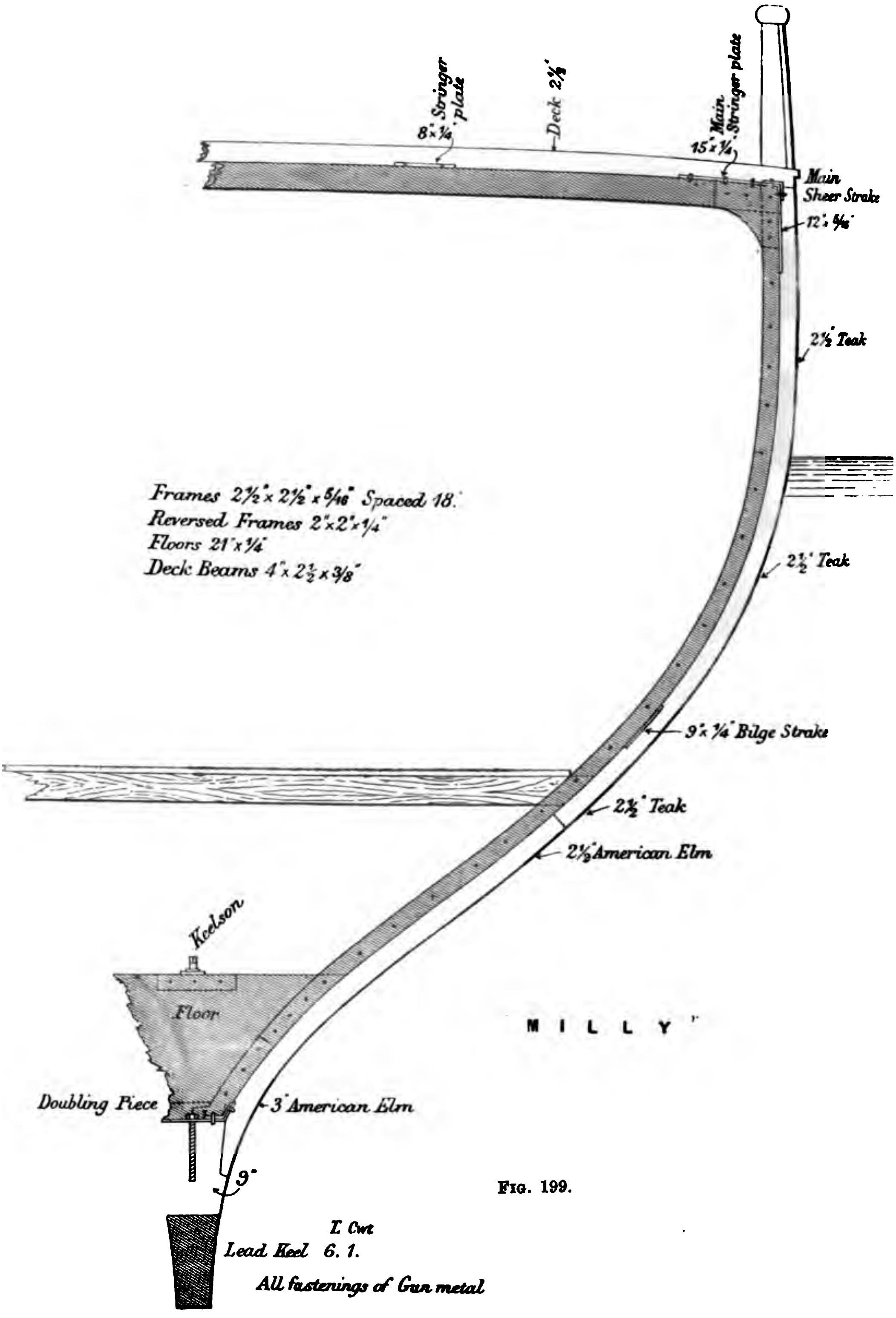


FIG. 198.



8" × 1/4" Stringer plate
Deck 2 1/8"
15" × 1/4" Main Stringer plate
Main Sheer Strake
12" × 5/16"
2 1/2" Teak
Frames 2 1/2" × 2 1/2" × 5/16" Spaced 18."
Reversed Frames 2" × 2" × 1/4"
Floors 21" × 1/4"
Deck Beams 4" × 2 1/2 × 3/8"
2 1/2" Teak
9" × 1/4" Bilge Strake
2 1/2" Teak
2 1/2" American Elm
Keelson
Floor
MILLY
Doubling Piece
3" American Elm
9"
T. Cwt
Lead Keel 6. 1.
All fastenings of Gun metal

Fig. 199.

constructed 40-tonner, with a broader wood keel, such as yachts at present are provided with. The iron keel plate is carried well up over the apron of stem, and to it the frames, &c., are riveted as shown. The keel plate is through bolted to the keel, as shown in plate representing the mid-section construction of Linotte and Tern, or fastened by a coach screw, as shown in Fig. 199. The garboard strake, which is fastened to the frame keel plate and keel, forms an additional attachment for binding the floor construction. However, even this construction, in the case of yachts with very heavy lead keels, has not been considered sufficient, and in building the Genesta, Mr. J. Beavor Webb introduced a steel garboard plate. In Fig. 200 this garboard plate is shown by *f*. Where the keel

KEEL CONSTRUCTION OF GENESTA.

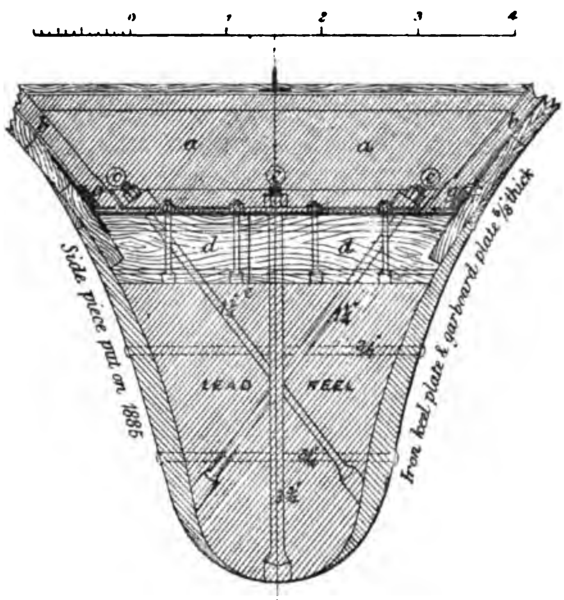


FIG. 200.

plate has not been flanged, an angle iron (*g*) has been worked (see also Plate XXI.). The floor plates (*a a*) are worked at every frame. The garboard plate is fastened to the frames (*b b*), angle iron (*g*), and bolted clean through the wood keel (*d d*). The wood keel is in two pieces, the seam being at *e*, and there is another bolt below, *d d*. The keel plate bolts are shown, also the lead keel boats. The horizontal bolts are for holding on the side pieces of lead which were put on in 1885. The small circles (*c c c*) are limber holes, the yacht being cemented up level with the angle floors, or heels of frames, which cross the top of the keel plate.

Lloyd's require that the plank shall be of the thickness of the wales

specified for wood yachts of similar tonnage. The plank is fastened to the frames by brass or gunmetal bolts, set up on the frames with nuts. The holes in the frames are punched or drilled, and after the plank is fitted on the frames and temporarily secured the plank is bored to receive the head of the bolt and its stem. The bolt is luted with white lead, and, with a thread of oakum under its shoulder, is punched in through the plank and frame and set up with a nut. A plug, also luted with white lead, is driven in over the head of the bolt to fill the hole in the plank. The thread of the bolt is opened to prevent the nut working off. From the bilge downwards, the frames and nuts are cemented over. The butts of the plank do not meet on a frame, but between the frames, and are secured by straps of iron or steel, the bolts and nuts being the same as the plank fastenings through the frames.

The yachts Linotte, Tern, and May, whose mid-sections are shown on Plates XX. and XXI., were assigned a 20 years class by Lloyd's. At this date (1890) the only other yachts which had received a similarly high class were the Marjorie and Lethe.

The 20 years class is made up as follows, according to Lloyd's rules (stem and sternpost only of English oak ; keel, English elm ; plank, teak and American elm) :

Wood material used	16 years
All outside plank fastenings yellow metal	3 years
Built under roof	1 year
	20 years

The years assigned to different timber by Lloyd's for composite yachts are as follow :

Description of Timber.	Keel.	Stem, Sternpost, Apron, Inner Sternpost, Deadwood, Knightheads, and Hawse Timbers.	Outside Planking. From top of Keel to two feet below Load-line.	Outside Planking. From two feet below Load-line to Planksheer.	Upper Deck Waterway, Covering-board, and Boughtree Timbers.	Rudder, Windlass, and Pallbitt. — Mainpieces.
East India Teak	16	16	16	16	16	16
Greenheart, Morra, and Iron Bark	14	12	14	12	12	14
English, African, French, Adriatic, Italian, Spanish, and Portuguese Oaks	14	12	12	12	12	14
Pitch Pine, Oregon and Huon Pine, Larch, Hackmatack, and Cowdie or Kaurie Pine	—	-	12	10	10	—
Northern Continental Oak	10	9	12	10	10	10
Dantzic, Memel, Riga, and American Red Pine	—	—	10	9	9	—
English Elm and American Rock Elm	16	—	16	—	—	—
Spruce Fir, Swedish and Norway Red Pine	—	—	8	8	—	—

IRON AND STEEL YACHTS.

Sailing yachts are occasionally built of iron or steel, and steam yachts are generally constructed of these materials. The advantage ascribed to the composite yachts, owing to the greater internal space compared with wood yachts for any given tonnage, is even more apparent in iron or steel yachts, as practically the thickness of the wood planking is available for increasing the internal breadth. The objections to iron or steel are that the variations of temperature in the outside air follow almost instantly inside, and it is therefore more difficult to keep an equable temperature in the cabins than it is in the case of a yacht planked with wood. Also the condensation of the humidity in the air is very rapid, and this may be an inconvenience both from its re-evaporation and from the damage it may do the plating and frames. Another objection is that the bottom is more prone to foul than that of a yacht covered with copper sheathing. However, the advantages due to the greater internal space and cheapness of construction more than counterbalance the objections, and, as far as yachts above 200 tons are concerned, steel is the best material for the frames and for the skin too, unless the yacht is destined to lie afloat for long periods together or make long voyages, in which cases the composite construction may be preferred.

It is beyond the scope of a work like this to give instruction in iron or steel yacht building, but it may be useful to quote a portion of Lloyd's Tables for steel yachts. (The whole rules are to be obtained at Lloyd's Register of Shipping, White Lion Court, Cornhill). For iron yachts (now seldom built) the sizes are about 25 per cent. larger.

Numbers for Frames, Reverse Frames, Floor Plates, and Bulkheads.	Spacing of Frames Centre to Centre.	Frames.		Reversed Frames.	Floor Plates.*		Engine and Collision Bulk-heads.
		Dimensions of Angles for three-fifths the length of vessel amidships.	Dimensions of Angles before and abaft the three-fifths lengths.	Dimension of Reversed Angles where required.	For Half-length amidships.	Thick-ness at Ends.	
	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.
20 and under 22·5	18	$1\frac{1}{2} \times 1\frac{1}{2} \times \frac{5}{20}$	$1\frac{1}{2} \times 1\frac{1}{2} \times \frac{5}{20}$	—	$7 \times \frac{5}{20}$	$\frac{5}{20}$	—
22·5 and under 25	18	$1\frac{3}{4} \times 1\frac{1}{2} \times \frac{5}{20}$	$1\frac{3}{4} \times 1\frac{1}{2} \times \frac{5}{20}$	—	$8 \times \frac{5}{20}$	$\frac{5}{20}$	—
25 and under 27·5	19	$2 \times 2 \times \frac{5}{20}$	$2 \times 2 \times \frac{5}{20}$	$1\frac{1}{2} \times 1\frac{1}{2} \times \frac{5}{20}$	$9 \times \frac{5}{20}$	$\frac{5}{20}$	—
27·5 and under 30	19	$2 \times 2 \times \frac{5}{20}$	$2 \times 2 \times \frac{5}{20}$	$1\frac{3}{4} \times 1\frac{3}{4} \times \frac{5}{20}$	$10 \times \frac{5}{20}$	$\frac{5}{20}$	—
30 and under 32·5	20	$2\frac{1}{4} \times 2\frac{1}{4} \times \frac{5}{20}$	$2\frac{1}{4} \times 2\frac{1}{4} \times \frac{5}{20}$	$2 \times 1\frac{3}{4} \times \frac{5}{20}$	$11 \times \frac{5}{20}$	$\frac{5}{20}$	$\frac{2}{16}$
32·5 and under 35	20	$2\frac{1}{2} \times 2\frac{1}{4} \times \frac{5}{20}$	$2\frac{1}{2} \times 2\frac{1}{4} \times \frac{5}{20}$	$2 \times 2 \times \frac{5}{20}$	$12 \times \frac{5}{20}$	$\frac{5}{20}$	$\frac{3}{16}$
35 and under 37·5	21	$2\frac{1}{2} \times 2\frac{1}{2} \times \frac{5}{20}$	$2\frac{1}{2} \times 2\frac{1}{2} \times \frac{5}{20}$	$2\frac{1}{4} \times 2 \times \frac{5}{20}$	$12 \times \frac{5}{20}$	$\frac{5}{20}$	$\frac{3}{16}$
37·5 and under 40	21	$2\frac{3}{4} \times 2\frac{1}{2} \times \frac{5}{20}$	$2\frac{3}{4} \times 2\frac{1}{2} \times \frac{5}{20}$	$2\frac{1}{4} \times 2\frac{1}{4} \times \frac{5}{20}$	$13 \times \frac{5}{20}$	$\frac{5}{20}$	$\frac{3}{16}$
40 and under 42·5	21	$8 \times 2\frac{1}{2} \times \frac{5}{20}$	$3 \times 2\frac{1}{2} \times \frac{5}{20}$	$2\frac{1}{2} \times 2\frac{1}{4} \times \frac{5}{20}$	$14 \times \frac{5}{20}$	$\frac{5}{20}$	$\frac{3}{16}$
42·5 and under 45	21	$3 \times 2\frac{1}{2} \times \frac{5}{20}$	$3 \times 2\frac{1}{2} \times \frac{5}{20}$	$2\frac{1}{2} \times 2\frac{1}{2} \times \frac{5}{20}$	$15 \times \frac{5}{20}$	$\frac{5}{20}$	$\frac{5}{20}$

MEM.—The numbers for regulating the sizes of the frames, reversed frames, and floors are obtained by the addition of half the greatest moulded breadth of the vessel, the depth at the middle of the length, from the upper part of keel to the top of the upper deck beams, with the girth of the half-frame section of the vessel at the same part measured from the centre line at top of keel to the upper deck stringer plate.

* Floor plates in engine and boiler space of steam yachts to be one-twentieth of an inch thicker than given in this table.

Numbers for Keel, Stem, Sternpost, and Plating.	Keel and Stem, and Sternpost for Sailing Yachts.	Stern Frames of Screw Yachts.	Thickness of Outside Plating, in 1-20th of an inch. Garboard Strakes, breadth and thickness. Half-length amidships.	Garboard Strakes, breadth and thickness. Ends.	From Garboard to the lower edge of Sheerstrake. Half-length amidships.	From Garboard to the lower edge of Sheerstrake. Ends.	Sheerstrake, breadth and thickness. Half-length amidships.	Sheerstrake, breadth and thickness. Ends.
	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.
under 1000	$4\frac{1}{2} \times \frac{5}{8}$	$4\frac{1}{2} \times 1\frac{1}{4}$	3	3	3	3	3	3
1000 and under 1300	$4\frac{3}{4} \times \frac{3}{4}$	$4\frac{3}{4} \times 1\frac{1}{2}$	4	4	3 and 4	3 and 4	4	4
1300 and under 1700	$5 \times \frac{7}{8}$	$5 \times 1\frac{3}{4}$	4	4	4	4	5	5
1700 and under 2200	$5\frac{1}{4} \times 1$	$5\frac{1}{4} \times 2$	5	5	4 and 5	4 and 5	6	5
2200 and under 2900	$5\frac{1}{2} \times 1\frac{1}{8}$	$5\frac{1}{2} \times 2\frac{1}{4}$	6	6	5	5	6	6
2900 and under 3900	$5\frac{3}{4} \times 1\frac{1}{8}$	$5\frac{3}{4} \times 2\frac{1}{4}$	6	6	5 and 6	5	6	6
3900 and under 5100	$6 \times 1\frac{1}{4}$	$6 \times 2\frac{1}{2}$	7	6	6	5	7	6
5100 and under 5800	$6 \times 1\frac{3}{8}$	$6 \times 2\frac{3}{4}$	7	6	6	5	8	7
5800 and under 6500	$6\frac{1}{4} \times 1\frac{1}{2}$	$6\frac{1}{4} \times 3$	8	7	6 and 7	5 and 6	8	7
6500 and under 7200	$6\frac{1}{4} \times 1\frac{5}{8}$	$6\frac{1}{4} \times 3\frac{1}{4}$	8	7	6 and 7	5 and 6	8	7

Mem.—The number for regulating the sizes of the keel, stem, sternpost, and the thickness of the outside plating is obtained by multiplying that which regulates the size of the frames, &c., by the length of the vessel.

Numbers for Keelsons, Stringers, Tie Plates, Rudders, and Decks.	Middle Line Keelson on Floors. Amidships.	Middle Line Keelson on Floors. Ends.	Keelson and Stringer Angle Bars.	Stringer Plate on Upper Deck Beams amidships.	Stringer Plate on Up. Deck Beams at Ends, and on Low. Decks Beam Ends, amidships.	Tie Plates on Beams.	Rudder. Diameter at Head.	Rudder. Diameter at Heel.	Thickness of Upper Deck.
	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	In.	In.	In.
under 1000		—	$2 \times 2 \times \frac{5}{20}$	$8 \times \frac{4}{20}$	$6 \times \frac{4}{20}$	—	$1\frac{1}{2}$	1	$1\frac{3}{4}$
1000 and under 1300	-	—	$2\frac{1}{2} \times 2\frac{1}{4} \times \frac{5}{20}$	$10 \times \frac{5}{20}$	$8 \times \frac{5}{20}$	$4 \times \frac{5}{20}$	2	$1\frac{1}{4}$	2
1300 and under 1700	—	—	$2\frac{1}{2} \times 2\frac{1}{2} \times \frac{5}{20}$	$12 \times \frac{5}{20}$	$9 \times \frac{5}{20}$	$5 \times \frac{5}{20}$	$2\frac{1}{4}$	$1\frac{3}{8}$	$2\frac{1}{4}$
1700 and under 2200	—	—	$2\frac{1}{4} \times 2\frac{1}{2} \times \frac{5}{20}$	$14 \times \frac{5}{20}$	$10 \times \frac{5}{20}$	$6 \times \frac{5}{20}$	$2\frac{1}{2}$	$1\frac{1}{2}$	$2\frac{1}{2}$
2200 and under 2900	Bulb. $6\frac{3}{4} \times \frac{6}{20}$	—	$2\frac{3}{4} \times 2\frac{1}{2} \times \frac{5}{20}$	$16 \times \frac{5}{20}$	$12 \times \frac{5}{20}$	$6 \times \frac{5}{20}$	$2\frac{3}{4}$	$1\frac{3}{4}$	$2\frac{3}{4}$
2900 and under 3900	Plate. $7\frac{1}{2} \times \frac{6}{20}$	$\frac{5}{20}$	$3 \times 2\frac{1}{2} \times \frac{5}{20}$	$20 \times \frac{5}{20}$	$15 \times \frac{5}{20}$	$6 \times \frac{5}{20}$	3	2	$2\frac{1}{2}$
3900 and under 5100	$8\frac{1}{2} \times \frac{7}{20}$	$\frac{6}{20}$	$3 \times 2\frac{1}{2} \times \frac{6}{20}$	$22 \times \frac{6}{20}$	$17 \times \frac{5}{20}$	$7 \times \frac{6}{20}$	$3\frac{1}{4}$	2	$2\frac{3}{4}$
5100 and under 6500	$9 \times \frac{8}{20}$	$\frac{7}{20}$	$3 \times 3 \times \frac{6}{20}$	$26 \times \frac{6}{20}$	$20 \times \frac{6}{20}$	$7 \times \frac{6}{20}$	$3\frac{1}{2}$	$2\frac{1}{8}$	$2\frac{3}{4}$
6500 and under 8000	$10\frac{1}{2} \times \frac{8}{20}$	$\frac{7}{20}$	$3\frac{1}{2} \times 3 \times \frac{6}{20}$	$28 \times \frac{7}{20}$	$21 \times \frac{6}{20}$	$7 \times \frac{7}{20}$	4	$2\frac{1}{4}$	3

Mem.—The number for regulating the sizes of the keelson and stringer plates, deck, &c., is obtained by multiplying that which regulates the size of the frames, &c., by the length of the vessel.

Size of Upper Deck Beams.				
Length of Beam Amidships.	Angle Bars.	Length of Beam Amidships.	Bulb.	Double Angle Bars.
Feet.	Inches.	Feet.	Inches.	Inches.
8	$2\frac{1}{2} \times 2 \times \frac{5}{20}$	20	$5 \times \frac{5}{20}$	$2 \times 2 \times \frac{5}{20}$
9	$2\frac{3}{4} \times 2 \times \frac{5}{20}$	21	$5 \times \frac{5}{20}$	$2\frac{1}{4} \times 2\frac{1}{4} \times \frac{5}{20}$
10	$3 \times 2 \times \frac{6}{20}$	22	$5\frac{1}{2} \times \frac{5}{20}$	$2\frac{1}{2} \times 2\frac{1}{4} \times \frac{5}{20}$
11	$3\frac{1}{4} \times 2\frac{1}{4} \times \frac{6}{20}$	23	$5\frac{1}{2} \times \frac{5}{20}$	$2\frac{1}{2} \times 2\frac{1}{2} \times \frac{5}{20}$
12	$3\frac{1}{2} \times 2\frac{1}{2} \times \frac{6}{20}$	24	$6 \times \frac{6}{20}$	$2\frac{1}{2} \times 2\frac{1}{2} \times \frac{5}{20}$
13	$3\frac{3}{4} \times 2\frac{3}{4} \times \frac{6}{20}$	25	$6 \times \frac{6}{20}$	$2\frac{1}{2} \times 2\frac{1}{2} \times \frac{5}{20}$
14	$4 \times 3 \times \frac{7}{20}$	26	$6\frac{1}{2} \times \frac{6}{20}$	$2\frac{1}{2} \times 2\frac{1}{2} \times \frac{6}{20}$
15	$4\frac{1}{4} \times 3 \times \frac{7}{20}$	28	$7 \times \frac{7}{20}$	$3 \times \quad \times \frac{6}{20}$
16	$4\frac{1}{2} \times 3 \times \frac{7}{20}$	30	$7\frac{1}{2} \times \frac{7}{20}$	$3 \times \quad \times \frac{6}{20}$
17	$4\frac{3}{4} \times 3 \times \frac{7}{20}$	32	$8 \times \frac{8}{20}$	$3 \times \quad \times \frac{6}{20}$
18	$5 \times 3 \times \frac{7}{20}$	—	—	—
19	$5\frac{1}{2} \times 3 \times \frac{7}{20}$	—	—	—

BOATS AND BOAT BUILDING.

There is no uniform practice in providing yachts with boats either as to number, length, breadth, or depth. Each builder or owner suits his judgment, fancy, or requirements. So far as number is concerned, yachts under 30 tons usually carry one boat only; from 30 tons to 80 tons two boats, and above 80 tons three boats.

For a yacht of 30 tons 14ft. is about the length of the boat. In yachts below 30 tons the length would be reduced down to the 9ft. dinghy, which a 10-tonner can carry.

A yacht of 40 tons would carry a 12ft. dinghy and 16ft. gig; 60 tons, a 13ft. dinghy and 18ft. gig; 80 tons, a 12ft. dinghy, 16ft. cutter, and 19ft. gig; 100 tons, a 12ft. dinghy, 17ft. cutter, 20ft gig; 120 tons, 13ft. dinghy, 18ft. cutter, 21ft. gig; 150 tons, 13ft. dinghy, 20ft. cutter, 23ft. gig; 200 tons, 14ft. dinghy, 22ft. cutter, 24ft. gig.

For the tonnage given, steam yachts usually carry boats somewhat longer than those named, as steam yachts are of greater length for their tonnage, and so have more room for the span required by the davits.

The dinghy should always be of good breadth and depth, with a flat floor, but there are no strict rules as to what the relative breadth and depth should be in yacht practice. The Admiralty, however, have rules for the sizes of boats as follow. [The depth is measured from the top of gunwale at the mid-length to the garboard strake inside at its junction with the rabbet in the keel]:

ADMIRALTY DIMENSIONS FOR BOATS (1890).

Description.	Length.	Breadth.	Depth.	No. of Oars.	Length of Oars.	Weight of Boat and Oars.	Will carry persons.
Dinghy	12ft.	5ft.	2ft. 1in.	4	11ft.	4cwt.	9
Dinghy	14ft	5ft. 2in.	2ft. 2in.	4	12ft. 6in.	5cwt.	10
Cutter	16ft.	5ft 7in.	2ft. 2in.	6	13ft.	7cwt. 3qrs.	14
Cutter	18ft.	6ft.	2ft. 2in.	6	13ft.	9cwt. 2qrs.	17
Cutter	20ft.	6ft. 4in.	2ft. 3in.	8	13ft.	11cwt.	21
Cutter	23ft.	6ft. 11in.	2ft. 4¼in.	8	13ft. 6in.	14cwt. 1qr.	27
Cutter	25ft	7ft. 3in.	2ft. 5¼in.	8	13ft. 6in.	16cwt. 1qr.	34
Cutter	28ft.	7ft. 6in.	2ft. 6¼in.	10	14ft.	18cwt. 1qr.	40
Cutter	30ft.	8ft. 1in.	2ft. 8¼in.	12	15ft.	19cwt. 2qrs.	50
Cutter-gig	20ft.	5ft. 6in.	2ft. 2in.	4	14ft.	7cwt. 1qr.	18
Gig	20ft.	5ft. 6in.	2ft. 2in.	4	15ft.	6cwt. 3qrs.	17
Gig	21ft.	5ft. 6in.	2ft. 2in.	4	15ft.	7cwt.	18
Gig	22ft.	5ft. 6in.	2ft. 2in.	4	15ft.	7cwt. 1qr.	19
Gig	23ft.	5ft. 6in	2ft. 2in.	4	16ft.	7cwt. 2qrs.	20
Gig	24ft.	5ft. 6in.	2ft. 2in.	[illegible]	16ft.	7cwt. 3qrs.	21
Gig	25ft.	5ft. 6in.	2ft. 2in.	5	17ft.	8cwt.	22
Gig	26ft.	5ft. 6in.	2ft. 2in.	5	17ft.	8cwt. 1qr.	23
Gig	27ft.	5ft. 6in.	2ft. 2in.	6	17ft.	8cwt. 2qrs.	24
Gig	28ft.	5ft. 6in.	2ft. 2in.	6	17ft.	8cwt. 3qrs.	25
Gig	30ft.	5ft. 6in.	2ft. 2in.	6	17ft.	9cwt. 2qrs.	26

It will be noted that the cutters are very large and powerful boats, much larger, in fact, than a yacht could carry. The dinghies are also larger, and fitted with thwarts for four oars. In the case of a yacht two men would be usually employed in a 14ft. dinghy, who would row an oar each, but in the smaller size one man only would be employed. In the case of the gigs it will be observed that the beam and depth, according to the practice in the Royal Navy, are uniform throughout.

As already stated, there is no uniform practice so far as yachts are concerned, but the following sizes will be found suitable :

DIMENSIONS FOR YACHTS' BOATS.

Description.	Length.	Breadth.	Depth.	No. of Oars.	Description.	Length.	Breadth.	Depth.	No. of Oars.
Dinghy ...	9ft.	4ft. 2in.	1ft. 7in.	2	Cutter......	26ft.	5ft. 8in.	2ft. 4in.	6
Dinghy ...	10ft.	4ft. 3in.	1ft. 7½in.	2	Gig	16ft.	4ft. 6in.	1ft. 9in.	2
Dinghy ...	11ft.	4ft. 4in.	1ft. 8in.	2	Gig	17ft.	4ft. 6in.	1ft. 9in.	2
Dinghy	12ft.	4ft. 4in.	1ft. 8½in.	2	Gig	18ft.	4ft. 6½in.	1ft. 9½in.	3
Dinghy ...	13ft.	4ft. 5in.	1ft. 9in.	2	Gig	19ft.	4ft. 7in.	1ft. 9½in.	3
Dinghy ..	14ft.	4ft. 6in.	1ft. 10in.	2	Gig	20ft.	4ft. 7½in.	1ft. 10in.	4
Cutter	15ft.	4ft. 8in.	1ft. 10½in.	2	Gig	21ft.	4ft. 8in.	1ft. 10in	4
Cutter......	16ft.	4ft. 10in.	1ft. 11in.	3	Gig	22ft.	4ft. 8½in.	1ft. 10½in.	4
Cutter......	17ft.	4ft. 11in.	1ft. 11½ n.	3	Gig	23ft.	4ft. 9in.	1ft. 10½in.	4
Cutter.. ...	18ft.	5ft.	2ft.	3	Gig	24ft.	4ft. 9½in.	1ft. 11in.	4
Cutter......	19ft.	5ft. 1in.	3ft. 0½in.	4	Gig	25ft.	4ft. 10in.	1ft. 11in.	4
Cutter.. ...	20ft.	5ft. 2in.	2ft. 1in.	4	Gig	26ft.	4ft. 11in.	1ft. 11½in.	4
Cutter......	21ft.	5ft. 3in.	2ft. 1½in.	4	Gig	27ft.	5ft.	1ft. 11½in.	5
Cutter......	22ft.	5ft. 4in.	2ft. 2in.	4	Gig	28ft.	5ft. 1in.	2ft.	6
Cutter.. ...	23ft.	5ft. 5in.	2ft. 2½in.	4	Gig	2 ft.	5ft. 2in.	2ft.	6
Cutter......	24ft.	5ft. 6in.	2ft. 3in.	4	Gig	30ft.	5ft. 3in.	2ft. 1in.	6
Cutter......	25ft.	5ft. 7in.	2ft. 3½in.	5	Gig	31ft.	5ft. 4in.	2ft. 2in.	6

Boats are not often built from lines or models, the boat builder trusting to his eye for making the boat of a suitable shape for her work. For sea work the floors of dinghies and cutters should be flat, and the greatest beam should run well fore and aft.

Plates IV., V., and VI. (Series D) give the lines of some well-formed boats built at Cowes for yachts.

It would be useless to attempt to give instruction as to the building of boats without moulds as usually practised, and what follows will consequently relate to building from moulds which have been fashioned from some design.

The operation of boat building from moulds is analogous to that already described. Stout moulds will be made and set up on the keel as shown on Figs. 201, 202, 203, and 204.

In the first place lay off the sections, keel and sternpost, and stem as described page 413. If the keel has a straight edge to top and bottom, it will not require to be laid off.

When the sections are laid off, proceed to make moulds to fit the

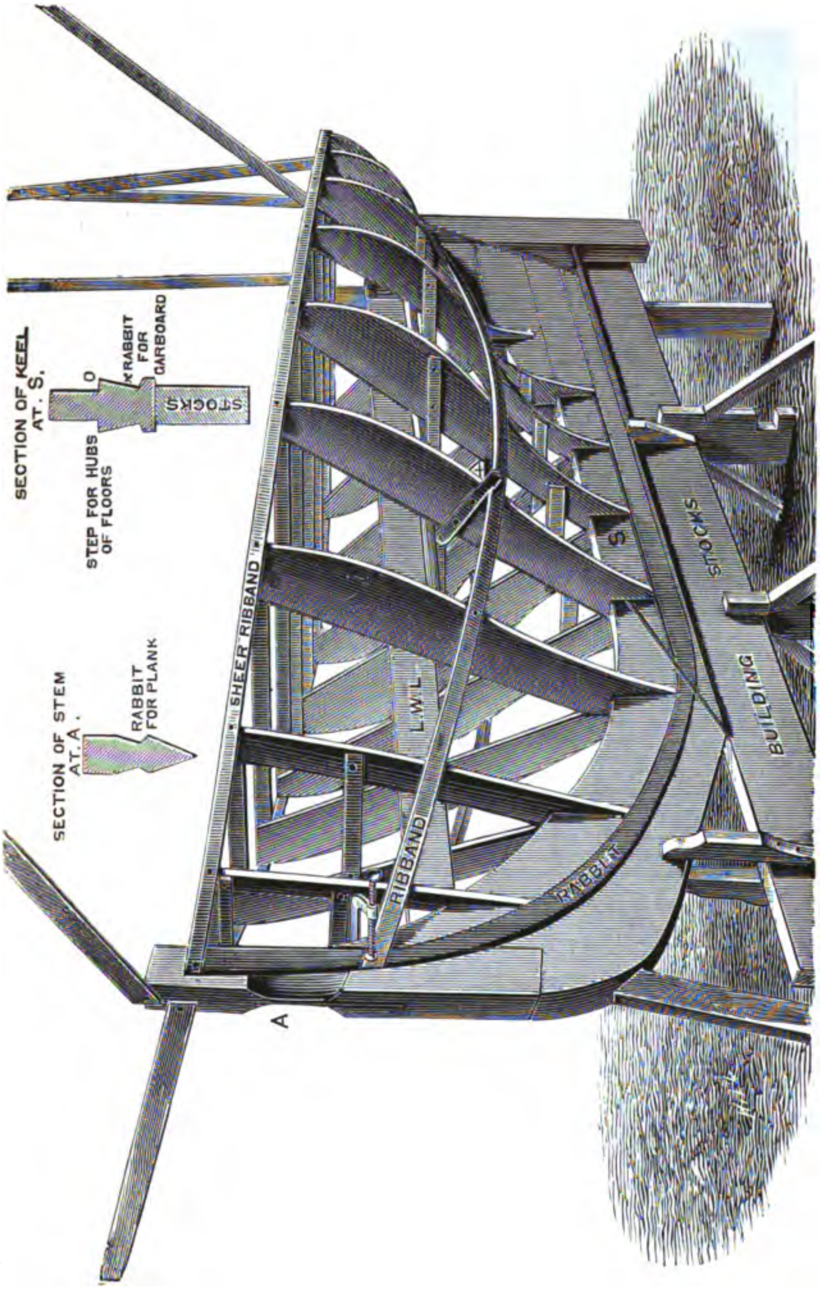

SECTION OF KEEL AT. S.
STEP FOR HUBS OF FLOORS
O
RABBIT FOR GARBOARD
STOCKS
SECTION OF STEM AT. A.
RABBIT FOR PLANK
SHEER RIBBAND
L.W.L.
RIBBAND
RABBIT
S
A
STOCKS
BUILDING

FIG. 201.

Fig. 202.

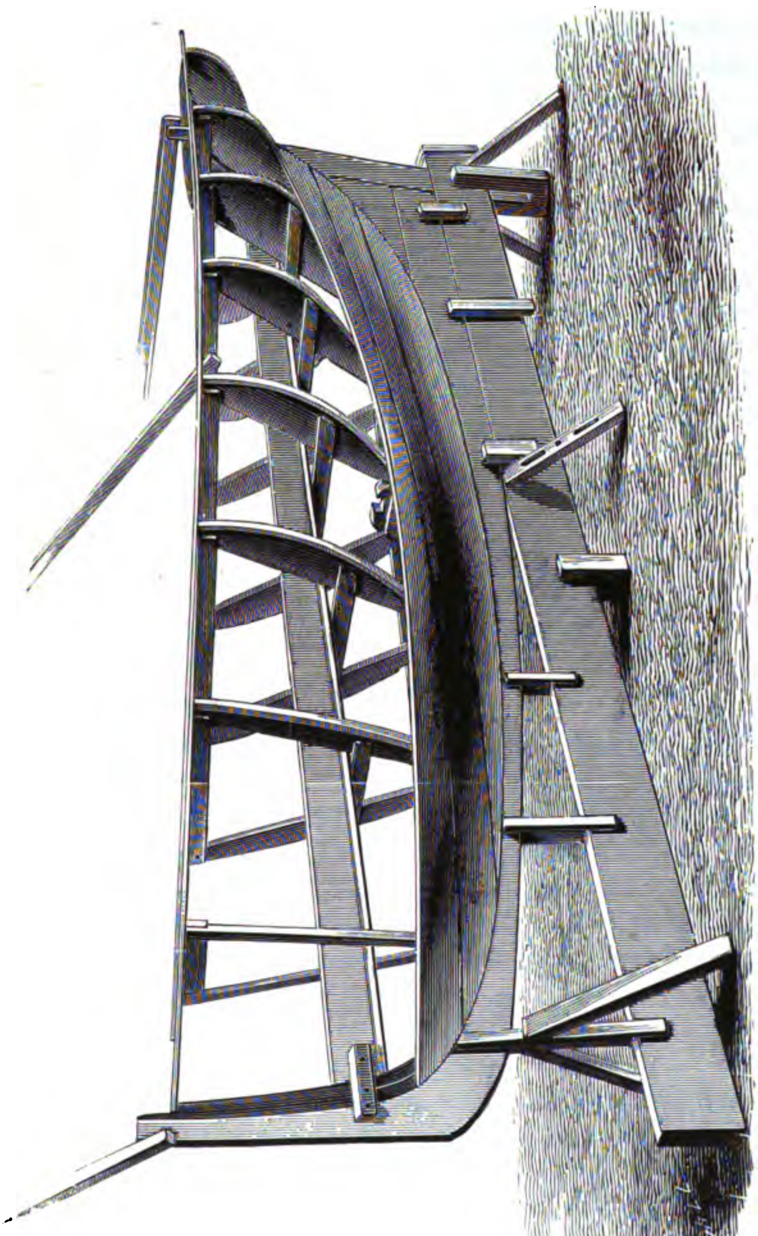


Fig. 203.

curves, these moulds would be made of ½in. or ⅝in. or 1in. deal or elm, according to the size of the boat; any odd pieces of stuff will do, and there can be as many joints in a mould as may be found convenient (see Fig. 204). The cross piece A should be stout enough to keep the mould rigid. The diagonal braces D need not be used if the mould can be made rigid without them; in such case the joints in the mould should be secured by a doubling piece. The bar W.L. represents the load water-line. B is the part that fits on the keel and represents the "jog" in the floor-timber. The cross piece here should be securely attached and fixed so that the jog is of the proper depth. A nail on each side of the mould, or a couple of pieces of wood nailed to the keel, will keep the mould in position on the keel.

The outer edges of the mould must be planed up to fit the curve of the section as drawn on the mould loft. When floors or timbers have to

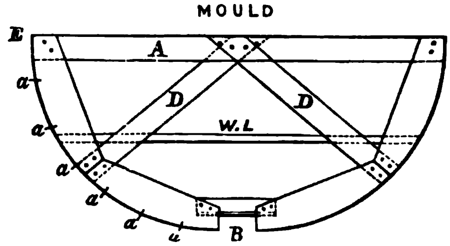


FIG. 204.

be sawn out, the mould is laid over the timber and its shape marked (see Fig. 175).

The stem and sternpost are tenoned into mortices in the keel; but if the keel be not thick enough to take a tenon, the keel and stem, and keel and sternpost are box-scarphed together by halves; that is, half the thickness of each is cut away.

The load water-line is made the base line, and therefore everything must be plumb to that.

Having got the stem, sternpost, and keel shaped and put together, proceed to fix them to the stocks. [A deal firmly fixed edgeways at a convenient distance from the floor—high enough to enable the builder to drive the nail up through the bottom of the boat—will make the stocks.] A straight edge or line must be fitted from stem to stern post to represent the load water-line as shown on the Sheer Plan. (See Fig. 201.)

Also a stout bar of wood must be nailed to stem-head and stern post-head, above the one marking the L.W.L., to firmly connect the two; this bar will be found useful for nailing the mould stays to.

In fixing the keel stem and stern post frame on the stocks it must be

wedged-up forward until the line or straight edge representing the load water-line is perfectly level or horizontal. A spirit level or plumb level can be used for this adjustment.

Fit the dead wood knee aft, and the stem knee or apron forward. Bore the holes for the through bolts with a long augur or gimlet. The heads of the bolts will be inside, and clenched outside over a ring. (See Fig. 205.)

Next the transom must be cut out from a mould and let into the stern post and through bolted, as shown in Fig. 205. The edges of the transom will require to be bevelled to suit the fore-and-aft curve of the boat.

When the keel, stem and stern posts are on the stocks and in position (the stem and stern post must be plumbed to see that they neither cant to port nor starboard), they must be secured by stays; the stays will be bars of wood and reach from the stem-head and head of stern post to the floor or rafters of the building shed, and they must be securely nailed. The keel can be kept in its position by similar stays; or if the keel be quite straight on its underside it can be kept in its position by thumb cleats nailed to the deal forming the stocks.

The lower edge and upper edge of the rabbet in the keel, stem, and stern dead woods must be next set off, and cut out with a chisel. The aft dead wood will probably require some adzing away back to the rabbet.

The moulds must be next put into their proper places. Care must be taken that they are "square" to the keel (*i.e.*, cross it at right angles) that they are plumb (perpendicular) to the load water-line, and that the bar W.L. (Fig. 204) is at the level of the line stretched between stem and stern post to represent the load water-line. The moulds must be kept in position by wood stays and ribbands formed by battens of fir. These ribbands can be let in flush with the outside edges of the moulds. They need not then be removed until the whole of the planking is complete and the timbers steamed in. This will be a great advantage for carvel build.

If the boat is to have a counter the rudder trunk will be constructed as explained on page 459.

In the case of clincher work the full width of the strakes (including the overlap of the lands) will be in the widest part about from 4in to 5in. Measure the half-girth of the midship mould (Fig. 204) from B to E by passing a tape or line round the outside curved edge. Divide this length into a number of equal intervals to represent the breadth of the strakes, as *a a*, &c. (see Fig. 204). Allowance must be made for about ¾in. overlap of each plank which forms the lands. Count the number of intervals or strakes of plank, and set off the same number in equal intervals on the

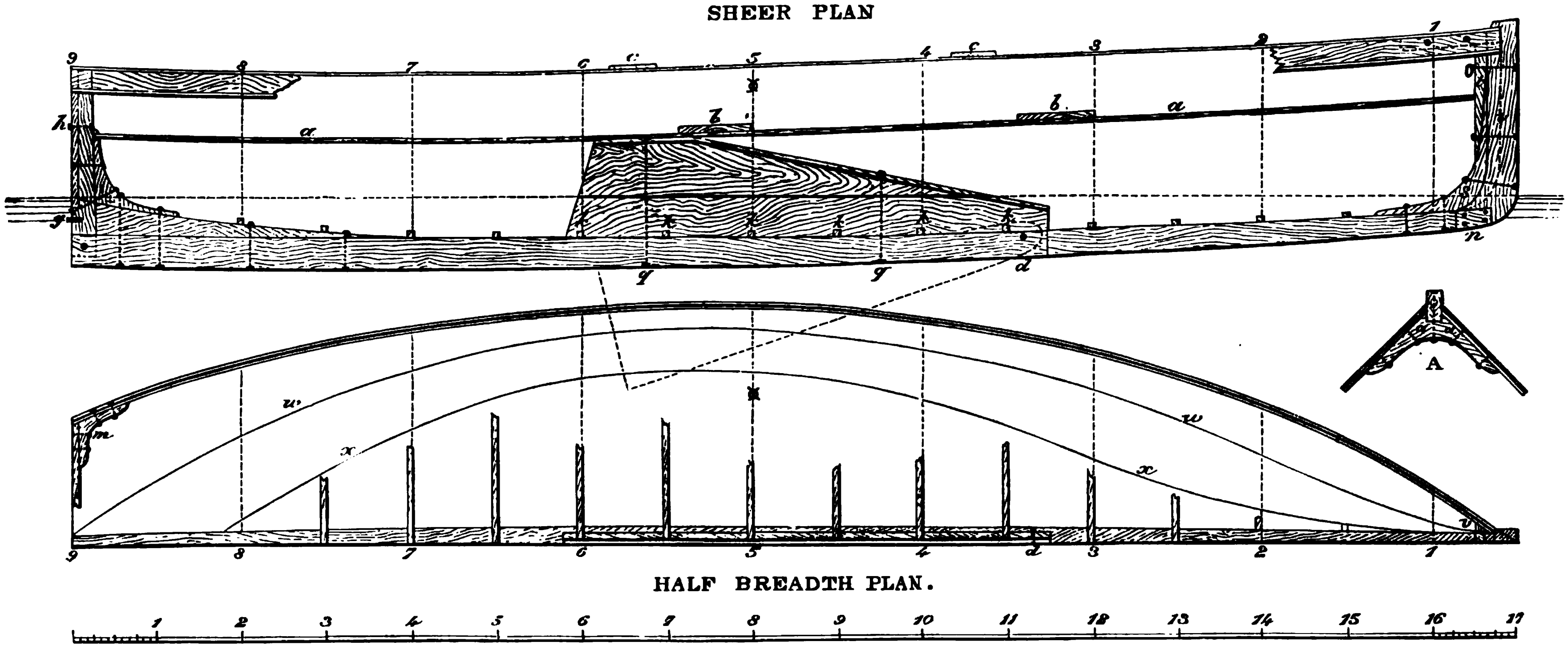


Fig. 205.

rabbet of the stem (see *x x*, Fig. 206) and on the transom. These intervals will be much closer together than on the moulds, and will therefore show that the plank must taper towards the ends. The same number of intervals can also be set off on the intermediate moulds.

The garboard strake will be first fitted. This will be a strake quite straight on its upper edge before it is bent round the moulds from stem to stern post. The under edge will be cut to fit the rabbet in keel, and stem, and dead wood aft. When this plank has been fitted into the garboard and nailed at intervals of two or three inches to the keel, the next plank must be fitted. Take the board (out of which the plank is to be sawn) and hold it along as closely as possible outside the upper edge of the garboard strake. Mark a line along it to correspond with the top edge of the garboard. Remove the board and it will be found that the line is more or less curved. Saw down this curved line. Then fit the board to the garboard again, making it overlap (by its curved edge) the garboard by about $\frac{3}{4}$ of an inch. Now mark by making spots on the upper edge of the board the

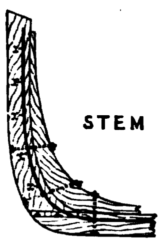


FIG. 206.

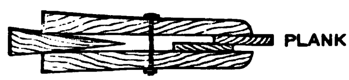


FIG. 207.

next interval on the moulds, representing the breadth of the plank (see *a a*, Fig. 204) for each mould, including stem and dead wood or transom aft. Remove the board and run a line in through the spots representing the intersections *a a* on the moulds (Fig. 204); this will show the shape, or the curve of the upper edge, and the curve of the lower edge of the next strake. The curve of the plank may be irregular, and the greatest curvature will be found as the bilge is reached. It is not absolutely necessary that the plank should accord with the spots *a a* at every mould, as the intervals are more as a guide to get the curve of the strake than to show the shape of the curve arbitrarily.

When the strake has been cut out it will be planed and bevelled, and then fitted to overlap the garboard; whilst it is being nailed it will be held in position by a number of clamps (at intervals of two or three feet).

The clamps are made of two pieces of hard wood loosely connected by a screw bolt (see Fig. 207) and has a wedge. The bolt must be allowed plenty of play, so that when the clamp grips the strakes it can be wedged up tightly as shown.

The plank will be nailed together at intervals of 3 inches. The nails will be of copper, and will be rooved and clinched inside. At the stem and transom, the upper part of each strake is thinned away in order that the hood ends may fit into the rabbet flush.

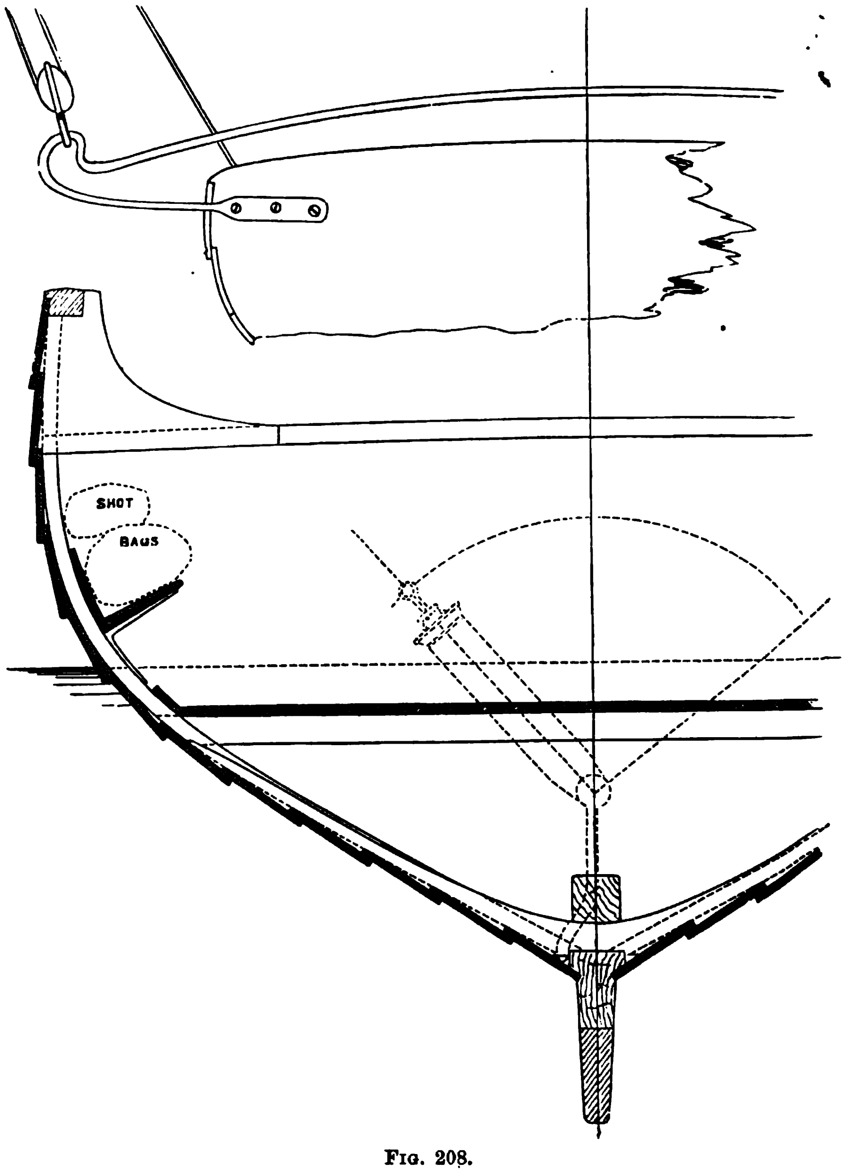


FIG. 208.

To get the strakes round the bilge in a fair curve the upper outer edge of each strake is bevelled off, and sometimes the inner lower edge of the overlapping strake is also bevelled.

Holes should be bored for the copper nails by a sharp bradawl a little

smaller than the nails. The roove or ring having been put over the nail, the latter will be cut off by a pair of nippers, leaving about $\frac{1}{16}$ of an inch projecting above the roove. A "holder-on" will be held to the head of the nail outside the boat, and the nail will then be clinched down over the roove inside the boat.

Fig. 208 shows a section of a Clyde clincher-built boat for racing, known as lugsail boats, and fitted with a ledge for shifting ballast, pump, inside kelson, &c.

The boat being planked up to the top strake, the floors and timbers must be put in. If the boat is a large one the floors will be sawn out of timber of a suitable size and jogged (see Fig. 209) to take the lands of the plank; but jogs are objectionable as they weaken the floors, and extra moulded depth should be given accordingly. The floors should extend across the keel and into the turn of the bilge; they will be fastened through the keel with a Muntz-metal, or copper, or galvanised iron bolt, and, if thought necessary, clenched on a ring. A copper nail will be driven through the plank (where two strakes overlap), and be clenched on the top of the floor; frequently rooves are not used for these fastenings. A fastening is put through every overlap.

The timbers should be sawn out of a straight-grained piece of American elm, but sometimes English oak or ash is used; oak is generally preferred for the floors, and American elm for the timbers.

The timbers, having been sawn out, must be planed up, and will then require steaming to get them into their places. The timbers should extend from one gunwale across the keel to the other gunwale, but frequently, where stout floors are inserted, the timbers are not worked across the keel, and do not reach within 6in. of the keel; in such cases the timbers are in "halves."

A steam chest or kiln will have to be constructed (see Fig. 210). In length it should be a foot or so longer than the longest timber, and be a foot deep and a foot broad. It should be made of 1½in. deal. The end, *a*, is a door. Inside on the bottom should be nailed three or four cross pieces of wood, 2in. deep, for the timbers to rest upon, forming a kind of ruck.

Steam can be generated in a common three-legged pot set up on bricks (see *k*). The pot should contain three or four gallons of water. The cover will be made of wood, cemented round with clay or mortar. *n* is a steampipe (made of inch deal, inside size 3½in. by 3½in); *p* is a plug for the water supply. To make steam for a chest to take long planks a larger boiler would be required, and a good contrivance is to rivet two galvanised iron "coppers" together by their rims, the top one of course being upside down, and thus forming a dome, with steam pipe *p*. (See Fig. 211.)

The door, *a*, in the steam chest should fit inside the steam chest, and fillets of inch deal must be nailed inside for the door to rest against. Before putting the door in its place, clay or mortar should be smeared round the fillets to keep the door from leaking. The door need not be hinged, but can be kept in its place by a cross-bar of wood working through two staples driven into the ends of the chest.

The timbers will require steaming three or four hours before they are sufficiently pliable. They must be taken from the steam chest and fitted into the boat one by one; the first fastening to put in will be the one through or in the keel (a Muntz-metal dump is best for this). Press with the foot or hands the timber into the bilge, and put a fastening through it here (from the outside). The stations for the timbers should be previously

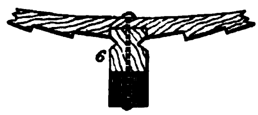
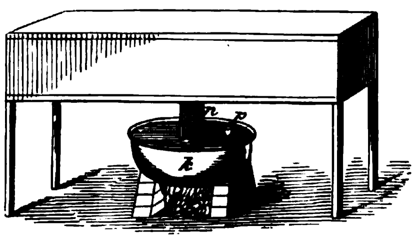
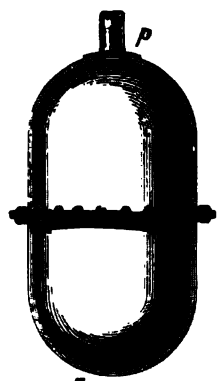


FIG. 203.



STEAM KILN

FIG. 210.



FURNACE

FIG. 211.

marked across each plank strake, and the holes through the overlaps should be bored before putting the timber in.

If the timber has to be jogged to receive the inside edges of each strake (see Fig. 209), the fastenings must not be clinched, as the timber will have to be removed for the jog to be cut. The timbers, however, should not be removed until they are perfectly cool and rigid; they should be allowed to stay in the boat a day and night before removing.

The gunwale must now be fitted (this is more properly termed the "in-wale," as it is the piece of timber which is fitted *inside* the top strake; it answers the purpose of the "clamp" used in larger boats). Having decided upon the size of the wale—its depth and thickness—it must be fitted. In the first place, the timber heads are cut down inside the top strake to the depth of the wale [one plan is not to cut the timbers so low as this by half an inch, and to make a jog in the wale to receive the head of each timber; when this is done, however, the wale or clamp should be somewhat stouter, as it will be weakened by the jog]. Usually the wale is flush with the top strake; but a better plan is to cut a rabbet in the

wale to fit over the top strake. A nail is put through the top strake and wale (from the outside), and rooved; or clinched without a roove, inside. A nail is put through about every 4, 5, or 6 inches. Forward, the wale, top strake, stem, and apron are kept together by a breast hook or > shaped knee (see sketch A, Fig. 205). Aft, the wale and top strake are secured to the transom by a knee (see *m*, Half-Breadth Plan, Fig. 205). The thwarts will rest on the stringers (which are fastened through timbers and plank), as shown by *a a* and *b b*, Sheer Plan, Fig. 205. The thwarts are secured by oak knees. The knee is fastened through and out the wale and top strake, and with a long fastening through the overlap of strakes, and clenched with ring on the knee; there will also be fastenings through the thwart and knee.

In buying copper nails care must be taken that "land nails" are obtained for the plank fastenings, and "timber nails" for the timber fastenings. The rooves must match the nails. A rooving iron (which is

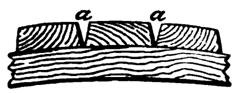


Fig. 212.

simply a kind of punch with a hole in its end) will be required to drive the rooves on whilst a hammer is held to the head of the nail. The sizes of the plank nails will depend upon the double thickness of the plank; about one-sixth of the double thickness should be added to the length of the nails for rooving and clenching.

If the boat is to be decked, a clamp or kind of shelf must be fitted to the timbers, and thoroughly fastened at each timber. The clamp will be fitted low enough for the beams to come flush with the top strake. The beams will be arched as required, and fastened through the shelf. The top strake should be of sufficient thickness to take the fastenings of the covering board. The covering board should be of hard wood, such as oak, and must be cut to fit the curve of the deck as shown in the half-breadth plan.

The deck plank will be nailed to the beams by galvanised nails; not *through* the plank from the top downwards, but diagonally through the side edges of the plank into the beams.

The under edges of the plank will meet closely on the beams; but the upper edges will "gape," as shown, in an exaggerated way, by *a a*, Fig. 212; this is for the caulking. An eighth of an inch or less will give a wide enough seam. The oakum or cotton thread (a couple of threads will be enough for each plank) must be driven in tightly by the caulking chisel, and then payed with marine glue or stopped with putty. Generally the arch of the beams will give the seam opening enough, as at *a a*; but

if the deck is quite flat, the plank should be bevelled to the extent of a shaving. The seam round the covering board should be caulked and payed with extra care. In case of match board decks, painted canvas is often laid over them. The deck is coated with thick paint, and the canvas stretched over it. The manner of fixing the canvas is shown on Plate XXVII. 2.

A hanging beam should be fitted on each side under the beam abreast of the mast or rigging. If the boat is wholly decked, three pairs of such hanging knees should be fitted.

If the boat is half decked, waterways should be fitted. Short beams will be worked for these, and their inner ends will be butted into a fore-and-aft beam termed a carline, which fore-and-aft piece will in turn be butted into the full beams at either end. Two or three pairs of hanging knees (made of oak) will support the waterways.

Fig. 205 shows a design for a centre-board gig which has been largely built from. The boats proved to be fast sailers, and very light to row. The drawing is made to half-inch scale; but, as it is rather small to work from, the following tables can be referred to in laying off.

The references to the body plan (Fig. 213) are as under: *w* is the load water-line (L.W.L.); *a a* 1 and *a a* 2; *b b* 1 and *b b* 2; *c c* 1 and *c c* 2 are "diagonals;" *o* is the middle vertical line, from which all distances are measured; *p p* are perpendiculars denoting the extreme breadth; *m m* is a kind of base line 10in. below the load water-line, and parallel thereto; *x* is a water-line.

The numerals 1, 2, 3, 4, &c., denote the respective sections or timbers and their stations in the sheer plan and half-breadth plan. No. 9 is the "transom," and of course will be a solid piece of wood, and not a "frame."

LAYING OFF TABLE.

Nos. of Sections.	1		2		3		4		5		6		7		8		9	
Sheer Plan and Half-breadth Plan.	ft.	in.	ft.	in.	ft.	in.	ft.	in.	ft.	in.	ft.	in.	ft.	in.	ft.	in.	ft.	in.
Heights above L.W.L. to top of gunwale	1	10¾	1	9¼	1	8¼	1	7¾	1	6¼	1	5¾	1	5¾	1	6	1	6½
Depths below load water-line to top of keel	0	2	0	8⅜	0	4¼	0	5	0	5¾	0	5¾	0	5½	0	5¼*	0	5
Depth of keel	0	3	0	3¼	0	3½	0	3⅞	0	4	0	4	0	4	0	4	0	4
Half breadths at gunwale	0	7	1	6½	2	2	2	7	2	9	2	8½	2	6	2	1	1	5½
Half breadths at L.W.L.	0	2¾	0	11¾	1	8½	2	3	2	5½	2	5¼	2	0½	1	3½	0	1¼
Body Plan.																		
Diagonal *a*	0	3¾	0	10¼	1	2¼	1	4½	1	6½	1	6¼	1	4¼	0	11¼	0	3
„ *b*	0	6½	1	6	2	1½	2	5½	2	7¾	2	7¼	2	4¼	1	10¾	1	0¼
„ *c*	0	9	1	11	2	7½	3	0½	3	2¾	3	2¼	2	11½	2	6	1	9

* Depth to top of dead wood, 3¾in.

The distance a (Fig. 213), above the load water-line w, is $3\frac{1}{2}$in. measured on the vertical line o. The distances a 1 and a 2 from the vertical line o, measured along the horizontal line m are 2ft. $3\frac{1}{2}$in.

The distance b, above the load water-line (w), is 1ft.; b cuts the perpendicular p at b 1 and b 2, 2in. below the load water-line w.

The distance c, above the load water-line w, is 2ft. 2in.; and c cuts the perpendicular p at c 1 and c 2, $3\frac{3}{4}$in. above the load water-line w.

x is a water-line struck 3in. below w, but will be of no assistance in laying off, as it does not intersect the frames sufficiently at right angles.

All the half-breadths, and the distances measured from the middle vertical line o along the diagonals to the various sections (as given in the tables), are *without* the plank; so in laying off *no allowance* will have to be made for the thickness of the plank.

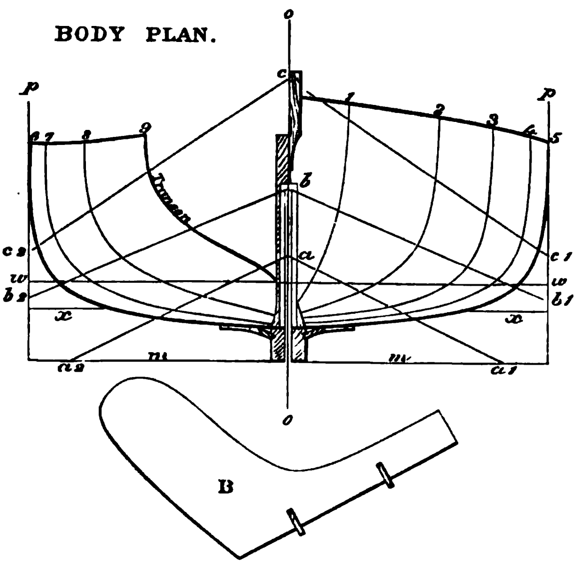


FIG. 213.

The length of the boat is 17ft., and the breadth 5ft. 6in., and the extreme breadth, with the plank on, 5ft. $7\frac{1}{2}$in. Weight of displacement of boat to L.W.L. about 12cwt.

The sections 1, 2, 3, 4, &c., are 2ft. apart, and No. 1 is 1ft. from the fore side of the stem at the L.W.L.

The frames actually will only be 1ft. apart; but every other one is left out in the Body Plan.

The scantling of the boat will be as follows: Keel, sided (thick) amidships $4\frac{1}{2}$in., tapered gradually to $2\frac{1}{2}$in. forward, and $3\frac{1}{2}$in. aft. The moulded depth of the keel will be found in the table.

Stem, 2½in. sided ; 4½in. moulded (*i.e,* its fore and aft thickness) at head, and 5½in. at knee scarph.

Sternpost, 3in. sided and moulded at heel ; 2in. moulded at head.

Floors, ¾in. sided and 2in. moulded (deep) at heels, gradually tapering to ¾in. at heads. The floors will be fitted across the keel by ½in. joggles. Timbers at the sides and above floors, ¾in. square.

Plank, ½in. thick. Gunwàles, 1in. thick, 1½in. deep.

Stringers (lettered *a* in the Sheer Plan) 1in. square, fastened through timber and plank. Seats and rowlocks as at *b b c c.*

The centre-board or plate will be 5ft. 4in. long, pinned or pivoted in the keel below the garboard strakes, as shown in the Sheer Plan and Half-breadth Plan (Fig. 205) at *d,* 5ft. 9in, from fore side of the stem at L.W.L. The slot in the keel to admit the plate will be 5ft. 7in. long by ⅜in. in width. The floors where this slot comes will have to be cut through. The heels of these floors will be fitted with ½in. joggles, and let in to the under side of the centre-plate case, as shown at *k k k k k k* in the Sheer Plan. One 4in. copper nail through each heel (outside the centre-plate case) will be sufficient to fasten these floors to the keel. (The case must be very carefully fitted to the shape of the keel, and luted with white lead.) The centre-plate case will be fastened through the keel by long galvanised iron (¼in.) bolts. The case will be made of inch pine ; the plates of ⅜in. iron.

The lands of a boat very much increase the resistance, and if the boat is intended for racing, she should be either carvel built and caulked, ribband carvel, or double skinned. The "double skin" plan is as follows : When the stem and stern posts have been set up and fastened off, and the building moulds carefully and strongly fixed in position, and firmly battened round at their heads by a kind of temporary gunwale, and the centre-board case or cases fitted and fixed, this framework is turned upside down, and again fixed in position. Thin, well-steamed planks of cedar, about "wager boat" thickness, are tacked in position edge to edge, over the moulds, as if for carvel-planking ; over this is then laid, plank by plank, a somewhat thicker skin of steamed cedar, the edges of which come over the centres of the planks of the inner skin. The two skins are then fastened off as if one, along the rabbet line, with brass screws ; the edges of the outer skin are then pierced along and copper nailed as in ordinary building. The craft is then turned up and the nails are clenched off on the inside. It will also be found necessary on some strakes to nail along the inner skin edges also. In such case the holes will be bored from inside and nails driven from outside. Very few timbers will be needed, and the double skin will be found to possess great strength. A

good coat of varnish or strips of varnished calico between the skins would no doubt add greatly to the strength and watertightness of the structure.

In the ribband-carvel build (see Fig. 214), the planks of, say, $\frac{3}{8}$in. to $\frac{1}{2}$in. stuff, are tacked on to the building moulds edge to edge. Ribbands of cleaned-grained oak, about $1\frac{1}{2}$in. wide and $\frac{1}{4}$in. or $\frac{3}{8}$in. thick, are laid along on the inside of the joints of the plank between the timbers which are placed in the vertical positions shown in the sheer plan; the edges of the planks are then pierced and nailed through the ribbands, and clenched on the inside, or they may be screwed into the ribbands. A stronger plan is to work a $\frac{3}{8}$in. ribband in whole lengths, cutting out

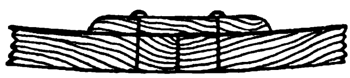


FIG. 214.

notches in the backs of the timbers and moulds to take each ribband. In all cases a strip of varnished linen should be laid over the joints of the plank before the batten is fitted, and the linen should be continuous from end to end. The timbers are about $\frac{1}{2}$in. sided by $\frac{3}{4}$in moulded. No doubt this mode, and that of the double skin, give a very fine outer surface; but the number of nails required is nearly double that employed in a clincher-built boat.

The garboard plank will be 6in. or 8in. wide at the broadest part, and the other planks will be as broad as the shape of the boat will admit of being worked, and will of course vary in breadth and shape.

The $\frac{1}{2}$-raters which are at present (1897) being built are planked with $\frac{1}{2}$in. cedar (sawn), which, when planed up, gives a thickness of $\frac{7}{16}$in. The timbers are single, of American elm, $\frac{3}{4}$in. sided, and $\frac{3}{8}$in. to $\frac{7}{16}$in. moulded, and regularly spaced centre to centre 6in. The edges of the plank fit closely, and are caulked and stopped. The lead, or fin bulb keels, weigh about 9cwt.

FITTINGS AND IRONWORK.

The mode of constructing the skylights, companion, hatches, &c., will be found in detail on Plates XXVI., &c. The most approved way of fitting these when the deck is $1\frac{3}{4}$in. thick, or more than $1\frac{3}{4}$in., is to let them in to the deck plank about $\frac{3}{8}$in. or $\frac{3}{4}$in. deep, with a luting of white lead on the bottom edges, and caulking and paying round the top of deck. The other plan is to fit the skylights, &c., close down on the beams and carlines. But the former plan is thought to give least trouble in the way of leaking should the beams work a little.

The fitting of cavels, cleats, belaying pins, bitts, capstan, &c., will be

found on Plates XV., XVII., XVIII. 4, &c. Internal fittings are delineated on Plates XVIII. 5, &c.

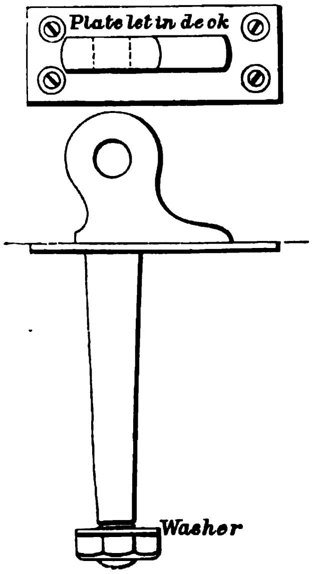


FIG. 215.

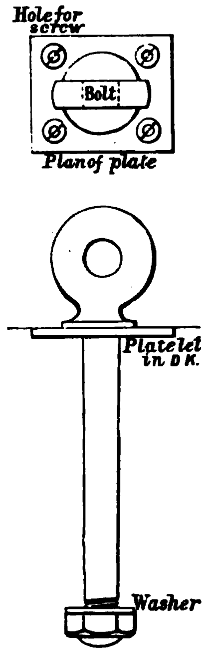


FIG. 216.

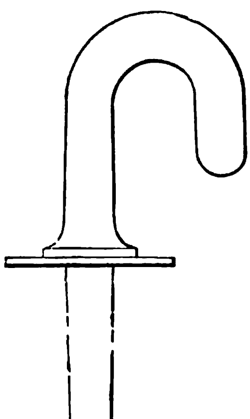


FIG. 217.

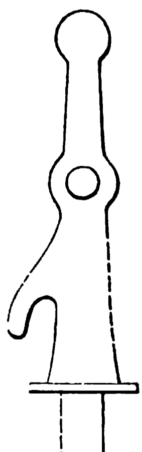


FIG. 218.

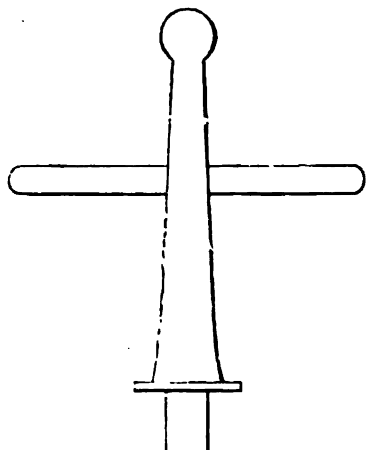


FIG. 219.

The ironwork about the deck and on the spars will be found on Plates referred to above. All the eye bolts in the deck for fore sheet, bowsprit shrouds, jib halyard †, &c., should go through a beam and be

set up with a nut and washer underneath, and square plate on deck. Sometimes the eye bolts, &c., have been screwed only into the deck and beam; but this is a most unsafe plan, and the bolts are almost certain to draw. All iron work should be galvanised.

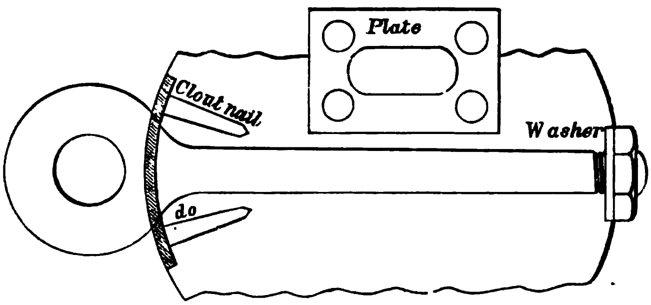


FIG. 220.

The bowsprit shroud bolts and trysail sheet bolts are usually similar in form and size, and have a "toe" as shown by Fig. 215.

If a beam does not come in the right place to take the bolts a chock should be worked underneath; and for bowsprit shrouds it is safer to work a piece of teak instead of the deck plank, as yellow pine has been known to crush through by the strain brought on the bowsprit shrouds.

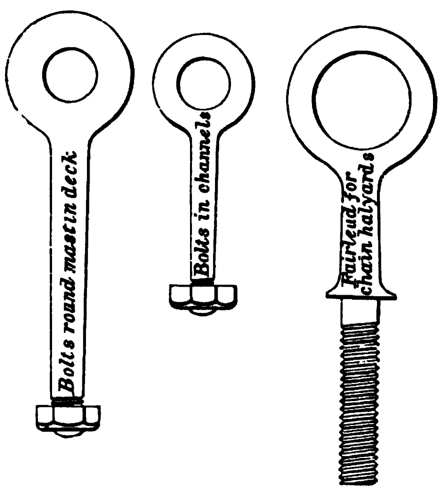
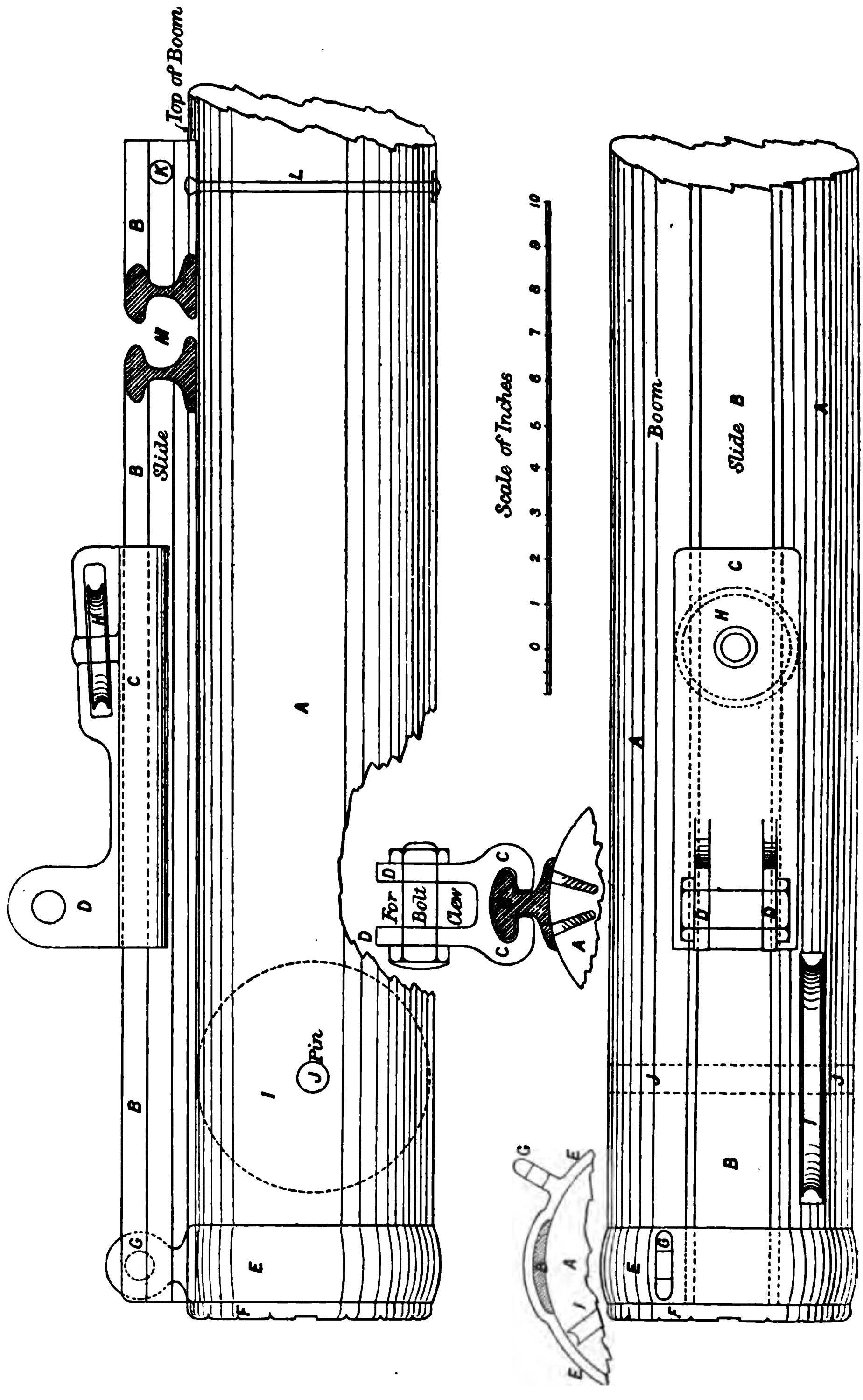


FIG. 221.

Fig. 216 shows fore-sheet bolt; two forms of bolts are used about the deck, as for foresheet, &c.

Fig. 217, jib chain halyard hook; Figs. 218 and 219, jib chain halyard cross.

The parts of the main halyard and peak halyard bolts which go through the mast taper somewhat, and are set up by a nut on a washer. On the front side there is a plate, and the eye of the bolt is sunk well in



Top of Boom
Slide
Scale of Inches
0 1 2 3 4 5 6 7 8 9 10
Boom
Slide B
For Bolt Clew
Pin

Fig. 212.

the mast, the slot in the plate being oblong for the purpose, as shown by Fig. 220.

Fig. 221 shows bolts round the mast and in the channel.

The fair leads on masthead for jib halyards, it will be seen by Fig. 221, screw in, as no great strain comes on them.

The fittings of the outer end of the boom are shown on Plate XXXII. Formerly the traveller was a leather bound iron ring and outhaul; then

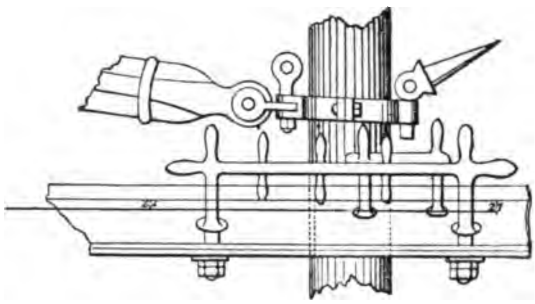


FIG. 223.

came the horse, as shown in the Plate; but since 1886 a slide has been adopted, as shown by Fig. 222. This slide, it will be observed, carries the clew of the main sail to the extreme boom end, a matter of importance for racing yachts. The inner end fittings and mast hoop of the main boom are shown on Plate XXXIII., and were taken from the original drawings for the Jullanar's ironwork. The spinnaker boom gooseneck, it will be seen, has a universal joint, and the boom ships and unships on the gooseneck, as shown by the dotted lines. It is, however, more usual to have a socket in

FIG. 224.

the mast hoop or gipsy winch band for the spinnaker boom gooseneck, as shown by Figs. 223 and 224.

The boom gooseneck and universal joint shown by Fig. 223 form the plan usually followed in small yachts; also the spinnaker boom gooseneck and joint. Sometimes, instead of a gooseneck, an iron socket is used, into which the end of spinnaker boom is shipped. In large vessels there is another mast band to take the gipsy winch; also the band is fitted with

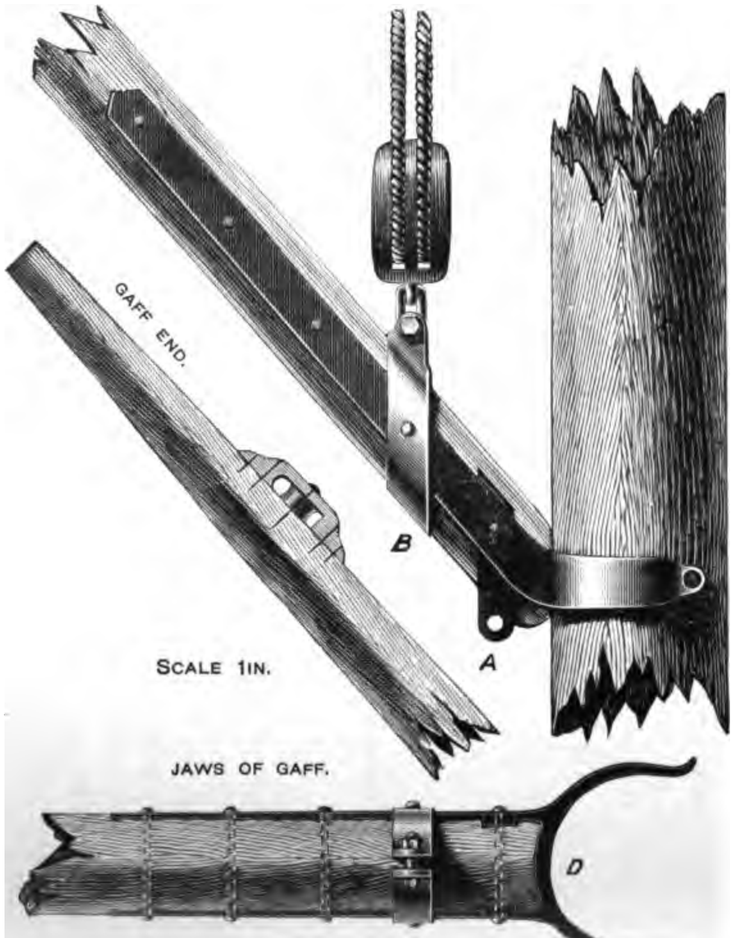


FIG. 225.

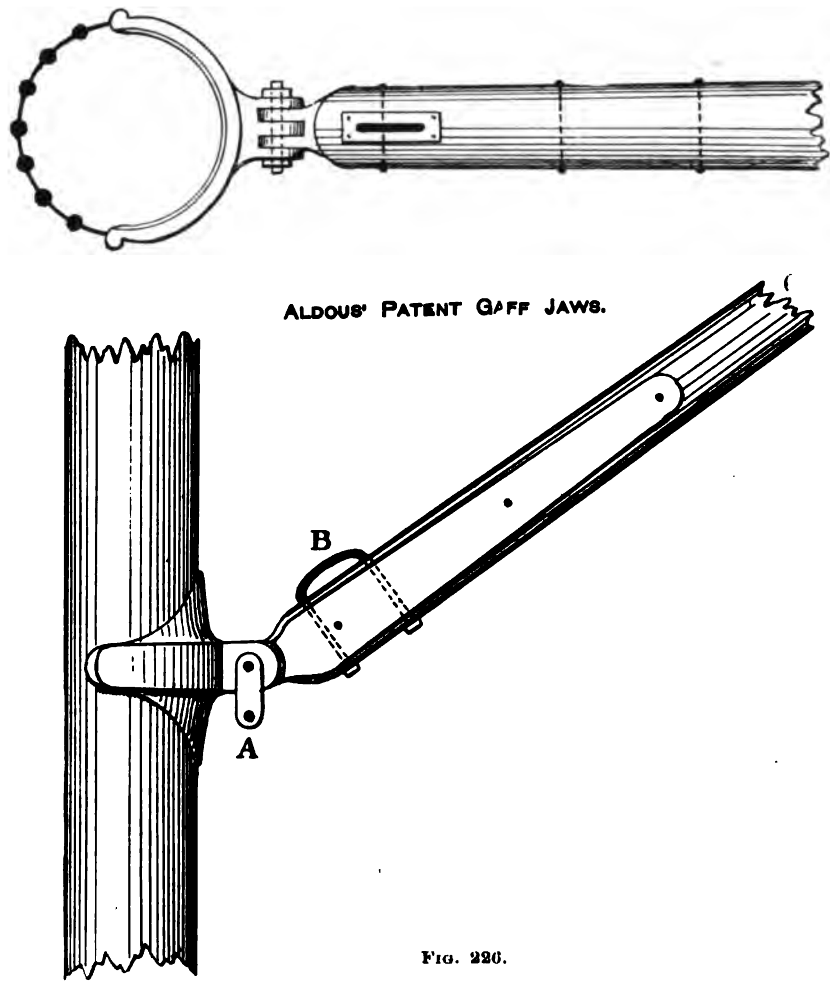


FIG. 226.

eyes to take belaying pins, hence it is called a spider band. In Fig. 224 the belaying pins are shown round the mast, also on deck, and under the bulwarks.

The jaws of a gaff are now generally made of iron of various patterns. A common plan is shown by Fig. 225, but occasionally wooden jaws are

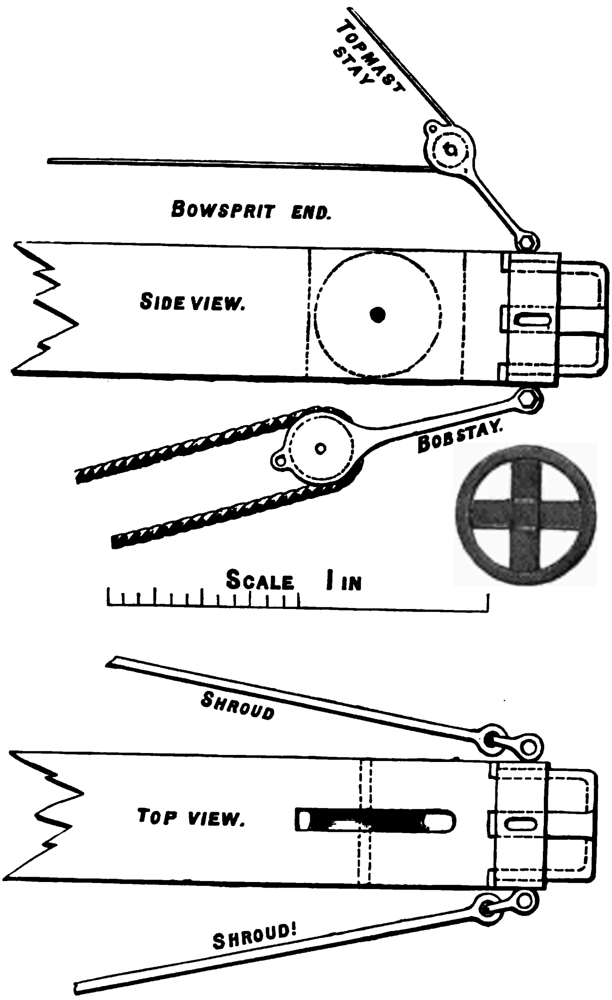


FIG. 227.

still met with. A represents a pair of eye-plates or lugs, to take the throat cringle or nock of the mainsail, a bolt going through all and set up with a nut. B is the throat halyard band; but it is quite as usual to have a slot in the gaff and a bolt through it, on which a long shackle works. However, in most racing vessels the plan shown is adopted, but the band is not always fitted in the oblique way shown

at B. One plan is not to bolt the arms of the jaws to the gaff, but to have a succession of iron bands fitted square to the gaff, and not diagonally, like B. Tumblers are dispensed with, D showing the inner part of the jaws, which are covered with thick hide. The outer end of gaff explains itself.

In 1876 Mr. Bentall adopted the "saddle" for gaff jaws, as shown in Plate XXXIII. Since then Mr. Aldous, of Brightlingsea, has simplified this plan, as shown in Fig. 216. In 1890 the saddle plan was further modified

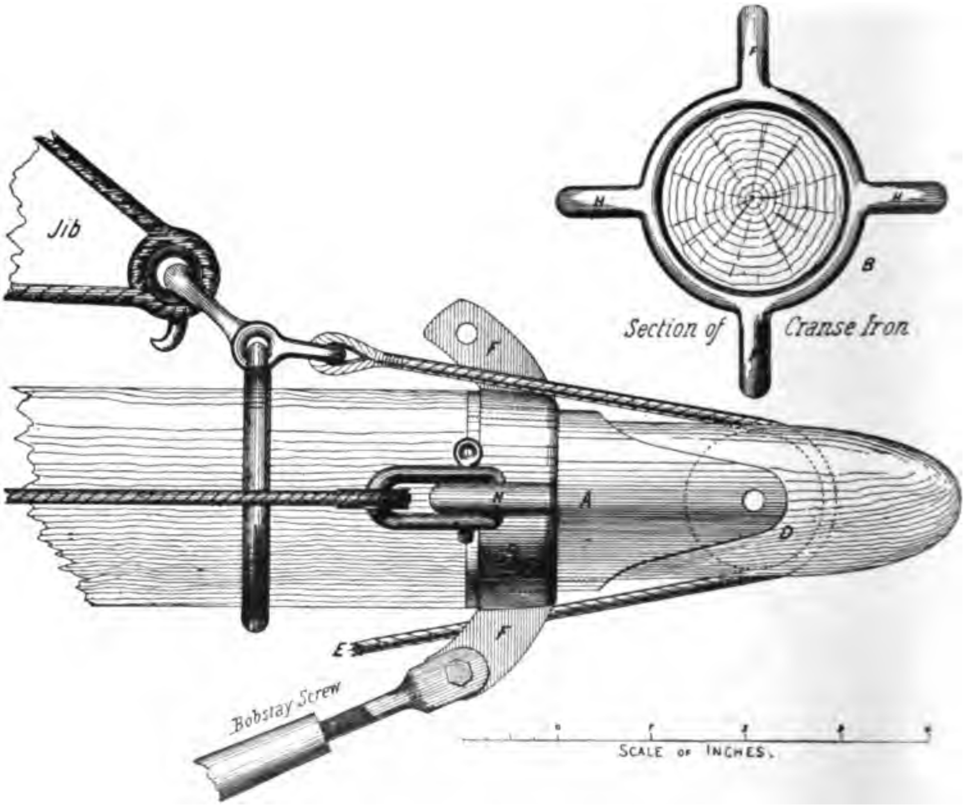


Fig. 228.

(as shown in Plate XXXIV.) by Mr. C. P. Clayton for the 20-rater Ghost, and manufactured by Messrs. W. White and Sons, Cowes.

The ironwork at the bowsprit end is delineated in Fig. 227 for a 40-tonner. The eyes to take the bobstay tackle, shrouds, and topmast stay must be very securely welded to the cranse iron. To prevent the cranse iron working into the wood an iron cross is fitted over the bowsprit end, and its ends turned up as shown.

It is obvious that the large score which has to be made to take the

sheave for jib outhaul and the bolt must greatly weaken the bowsprit end; and a plan some time ago introduced for putting the sheave outside the cranse iron, now generally adopted in small racing vessels, is shown by Fig. 228, in which it will be seen the sheave for jib outhaul is outside the cranse iron.

The sheave D is put a little out of the vertical so as to allow the outhaul to pass clear of the lugs F F.

There appears to be no fixed rule for determining the size of the iron fittings for deck and spars, but the following table has been compiled from general practice:

TABLE OF SIZES FOR IRONWORK FOR RACING YACHTS.

	3 Tons.	5 Tons.	10 Tons.	20 Tons.	40 Tons.	60 Tons.	100 Tons.	200 Tons.
	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.
Mast hoops or spider band for main boom and winches	1½ × ⅜	1⅝ × ⅜	1¾ × ⅜	2¼ × ½	2¾ × 9/16	3¼ × ⅝	4 × ¾	5¾ × 1
„ head cap	2 × ⅜	2¼ × ⅜	2½ × ⅜	2¾ × ⅜	3 × ⅜	3¾ × ½	4½ × ½	5½ × 1
„ centre connection	1⅜	1½	1⅝	1¾	2	2½	2¾	3
„ throat halyard bolt outside of and through mast	¾ & ½	⅞ & ½	1⅛ & ⅝	1 & ¾	1⅜ & 1	1½ & 1⅛	1¾ & 1¼	2½ × 1¾
„ peak halyard bolts	½	⅝	¾	⅞	1	1¼	1½	2
„ jib halyard band and eyes	2¾ × ⅜	3 × ⅜	3¼ × ⅜	3½ × ½	4 × ½	4½ × ⅝	5 × ¾	5¾ × 1
„ topping lift eye plates	1⅞ × ⅜	2 × ⅜	2½ × ⅜	2¾ × ½	3 × ½	3¼ × ⅝	3½ × ¾	4 × 1
„ fair lead for jib halyards	½	½	½	⅝	¾	1	1¼	1¾
„ eye bolts in deck round mast	5/16	½	9/16	⅝	¾	1	1¼	¾
Chain plates at the angle round the channel	1¾ × 1	1⅞ × 1	2 × 1⅛	2¼ × 1¼	2¾ × 1½	3 × 1⅝	3¼ × 1¾	4 × 2
„ below the channel (Plate XXVIII.)*	1¾ × ⅜	1⅞ × ⅝	2 × ½	2¼ × ½	2¾ × ⅝	3 × 11/16	3¼ × ¾	4 × 2
„ eye bolts on top of channel plate	⅜	⅜	⅜	7/16	½	⅝	¾	1
Main boom: iron at neck of universal joint and along boom	1 & 5/16	1½ & ⅜	1⅞ & ⅜	2 & ½	2¼ & 9/16	2½ & ⅝	3 & ¾	4 × 1¼
„ ears of universal joint	1¾ × 5/16	2 × ½	2¼ × ⅝	2½ × ¾	3 × ⅞	3½ × 1	4 × 1¼	5 × 1½
„ band for topping lifts	1½ × ⅜	1¾ × ⅜	2 × ½	2½ × ⅝	3 × 11/16	3½ × ¾	4 × ¾	5 × 1¼
„ travelling band for mainsheet	1 × ⅜	1 × ⅜	1¼ × ½	1½ × ½	1⅝ × ⅝	1¾ × ⅝	2 × ¾	2¾ × 1½
„ lugs for mainsheet	½	⅝	¾	⅞	1	1⅛	1¼	2
„ horse for clew traveller	¾	¾	⅞	1	1¼	1½	1¾	2½
Gaff: jaws round the mast	2 × ⅝	2 × ¾	2½ × ⅞	2⅝ × 1	2¾ × 1⅛	3 × 1¼	3¾ × 1¼	5 × 2½
„ in the angle at gaff end	2¼ × 1	2¼ × 1	2¾ × 1	3 × 1⅜	3 × 1½	3½ × 1¾	4 × 2	5 × 2½
„ throat halyard bolt and shackle	¾ & 7/16	¾ & 7/16	⅞ & ½	1 & ⅝	1⅛ & ¾	1¼ & 1	1½ & 1¼	2 × 2
„ eye plates for throat cringle of sail	⅜	7/16	7/16	½	⅝	11/16	¾	1¼
Bowsprit traveller	½	½	⅝	¾	1	1⅛	1¼	2
„ span shackle	2 × ⅝	2⅛ × ⅝	2¼ × ⅝	2½ × ⅝	2¾ × ¾	3 × 1	3¾ × 1¼	4¼ × 1^{3}
„ cranse	2 × ½	2¼ × ½	2½ × ⅝	2¾ × ⅝	3 × ¾	3½ × ⅞	4 × 1	5 × 1
„ cranse cross irons	1 × ¾	1 × ⅜	1⅛ × ½	1¼ × ½	1½ × ½	1¾ × ½	2 × ¾	2½ × 1
Mainsheet bolts in deck	¾	⅞	⅞	1	1¼	1½	1⅜	2
Trysail sheet bolts	¾	¾	⅞	1	1⅛	1¼	1⅜	2
Fore sheet bolts	9/16	⅝	11/16	¾	⅞	1	1⅛	1⅜
Bowsprit shroud bolts	¾	¾	⅞	1	1⅛	1¼	1⅜	1¾
Jib halyard + or hook in deck at mast	½	9/16	⅝	¾	1	1¼	1½	2
Eye bolts round the mast	½	½	9/16	⅝	¾	⅞	1	1½
Runner eye plates on the stanchions	1¼ × ½	1¼ × ½	1⅜ × ½	1½ × ½	1¾ × 9/16	2 × ⅝	2½ × ¾	3 × 1

* For sizes of rigging screws see Table I.

In racing yachts rigging screws are now being generally adopted, and their sizes will be set forth in Table I. The long sleeve nut has a right and left handed Whitworth thread, and is usually made of gun-metal. The screws are made of steel. The pattern of these rigging screws is shown on Plate XXIX. The pattern for small yachts of 5-rating and under is shown by Fig. 229. The lugs, A, takes a solid thimble in eye splice of rigging. If screws are used for the bowsprit shrouds, they would be of the size of those for the main rigging. (See Plate XVII., showing the deck plan of a 20-rater.) For most yachts a capstan is fitted on the fore deck instead of a windlass, as made by W. White and Sons, of Cowes, or Cantelo, of Southampton. For small vessels most of the yacht fitters make a small capstan with winch top, or a ratchet windlass. One of the latest of the latter is shown on Plate XXXV. It was planned by Mr. C. P. Clayton for the 20-rater Ghost, and made by Messrs. W. White and Sons, Cowes.

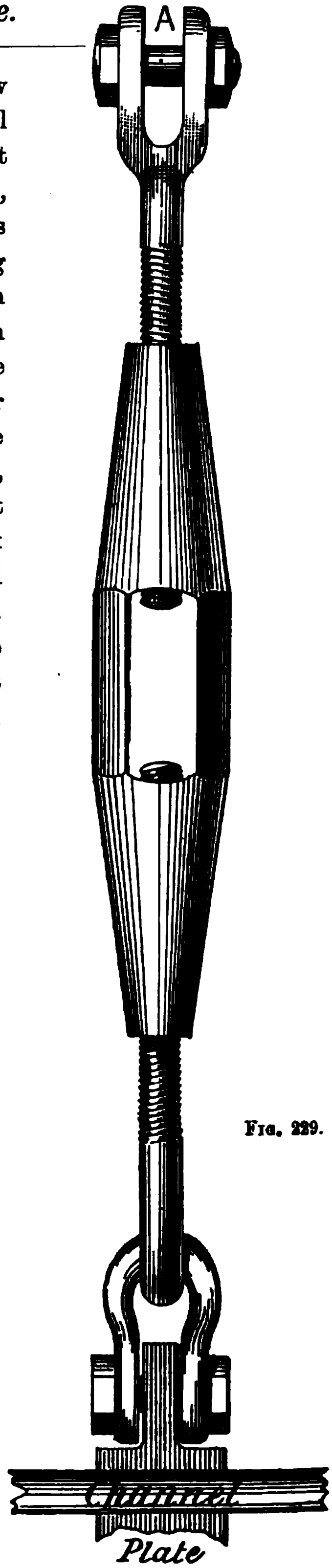


FIG. 229.

CHAPTER XVIII.

SPARS.

A GREAT many rules have been from time to time devised for the placing of masts in yachts, and for regulating their lengths; but, as there appears to have been an inability to base these rules upon the actual stability of a vessel, or upon the sail area she could effectively carry, they are little better than guesswork, and a yacht is generally sparred upon what may be termed a principle of comparison. The designer, according to his judgment and experience (which generally lead him to a right, although occasionally to a wrong, result), fixes upon the length of the spars without any reference to rules whatever, as he is conscious that the rules, contrived on general principles, might lead him astray. The result of the chance system is that the masts very frequently have to be reduced or increased in length.

No doubt masts affect the stability of a vessel to an enormous extent, and we find, from the calculations made for the Seabelle, on page 378, that 30 per cent. of her ballast as stowed goes to counteract the weight of her masts alone. A cutter is better off in this respect, as she has only one mast; but even in a cutter as much as 20 per cent. of the ballast is found to be required to balance the moment of the mast, and 25 per cent. of all the spars. As masts so largely affect the stability of a vessel, it will perhaps be conceded that a little more care and trouble should be taken in deciding upon their lengths. The position of a vessel's centre of gravity can be approximately calculated before she is built, and the exact influence the mast will have on that centre will form a large item in the calculation. Troublesome as the calculations may be, the trouble is small compared with what often has to be done when a mistake has been made in fitting a vessel with spars.

If the sail area is predetermined by area of wetted surface or displacement $\frac{2}{3}$ (see page 101, &c.), it would be a simple matter to compare the spars of well-known vessels with the square roots of the sail areas as a guide. In practice it is found that, as a rule, the main boom is longer in proportion to the square root of the area than masts, for the obvious

object of making the mast as light as possible and keeping the centre of effort low.

LENGTH ON SPARS FOR SCHOONERS.

An old rule for schooners was, in the days when yachts were about three beams to length, with inside ballast of iron only, as follows: Foremast, measured from deck to hounds, equal to the breadth multiplied by 2·7, and length of the masthead equal to 0·18 the length of mast from the deck to the hounds.

Mainmast, equal to the breadth × 2·85, with length of masthead equal to 0·18 of the length of mast from deck to hounds. This rule still holds fairly good for cruising vessels, but is insufficient for racers. Thus the Miranda has a mainmast deck to hounds 3·12 her extreme beam, and she is by no means a narrow vessel. The length of mainmast deck to hounds of a modern racing schooner is usually $\sqrt{\text{mainsail area}} \times \cdot 95$; and in a cruising schooner 0·85. This, of course, assumes that the figure of the sail is predetermined.

Main boom multiplied by 0·58 of the length on the water-line for cruisers, and 0·75 for racers, and main gaff in length from ·57 to ·61 of the length of the boom. So far as modern practice goes, we find that the length of main booms varies as the square root of the mainsail area × 0.85 to 0·90. The latter ratio more often obtains in cruisers because they have less comparative hoist.

Length of fore gaff equal to 0·322 of the length on the load water-line; but generally the fore gaff is made 0·8 of the distance the masts are apart.

Bowsprit (or bowsprit and jib boom taken together) outside the stem, equal to breadth × 1·75. But in almost all descriptions of rig the length of bowsprit is governed by the requirements of the centre of effort in relation to the centre of lateral resistance, due care always being exercised so that the bowsprit is not longer or heavier than necessary.

Fore spinnaker boom equal to the length from foremast to bowsprit end, because fore spinnaker was often used as bowsprit spinnaker, and is taxed if it exceeds that distance by the rating rule. Fully equipped racing schooners, carry three spinnakers—main, fore, and bowsprit.

Main spinnaker boom equal to length of mainmast deck to cap × 1·2; but generally the main spinnaker boom is made so that it will just clear under the triatic stay without its being unshipped at the gooseneck.

The length of main topmast from fid to hounds, or to sheave hole for topsail halyards, ·65 of the length of mainmast deck to hounds; the foretopmast, ·9 of the length of maintopmast.

Working gaff topsail yard, equal to the length of the gaff over which it is set.

Length of Spars for Cutters.

For cutters, the length of mast, deck to hounds, was formerly put at three times the greatest beam; but this rule is never now used, and modern practice shows the proportion to be approximately as $\sqrt{\text{mainsail area}} \times \cdot 9$; but in yachts of 52ft. rating the height deck to hounds is sometimes equal to $\sqrt{\text{mainsail area}} \times \cdot 95$. In yachts of 42ft. rating and under the height deck to hounds varies from 0·9 to 0·95. Masthead 0·24 of the length deck to hounds.

Main boom, equal to L.W.L. $\sqrt{\text{mainsail area}} \times 1{\cdot}1$ to $1{\cdot}15$ in small.

Main gaff, 0·6 to 0·63 of main boom. Trysail gaff, ·5 main gaff.

Angle of gaff with horizon, 50° to 55°.* As before said, this assumes the figure of the sail to be predetermined according to the present fashion.

The eye of the throat halyard bolt usually comes about ·04 the height deck to hounds above the underside of yoke $= \frac{1}{25}$. The jaws of gaff about ·062 $(= \frac{1}{16})$ below the yoke. This allows ample drift between the blocks so as not to get an unfair strain as the gaff goes off to leeward.

Bowsprit outside the stem, from 0·25 to 0·4 of the length on the load water-line.

Length of bowsprit housed is usually the length outside × ·4.

The topmast of cutters in length, from fid to hounds, was formerly ·8 of the length of the lower mast deck to hounds. In small cutters of 40 tons and under, the ratio was frequently ·9, and occasionally a 10-tonner was met with the length to topmast equal to length of lower mast.

These proportions have been altered since 1887, when the rating rule by sail area and length came into operation, the length of topmast having been considerably decreased in small vessels. Thus in a 52ft. rater or 42ft. rater the topmast is now about ·8 of the height deck to hounds. For the masthead fittings of a 52ft. rater see Plate X. 3 (A). For the masthead fittings of a 36ft. rating yacht see Plate XXXI.

Topsail yards vary in length a great deal, but to some extent are guided by the length of the gaff over which they are set. Formerly all racing vessels carried a "balloon topsail," or a topsail with a long head yard, whilst the foot was extended beyond the gaff by a foot yard or jack yard. The head yard of a balloon-topsail was from 1·3 to 1·5 the length of the gaff over which it was set. The modern practice is to have a very much shorter yard than this, and to make the sail longer in the luff. The

* Twenty years ago sails were cut much flatter in the head than at present; but a sail with a high peak appears to stand as well as one with a low peak, and a much larger sail for the same weight of spars can be obtained by increasing the peak.

No. 1, or working topsail,* had a yard equal to the length of the gaff, and No. 2 topsail a yard equal to ·6 of the length of the gaff.

Spinnaker booms vary in length from 1·1 to 1·3 the length of lower mast deck to hounds, or it may be governed by the distance from the mast to the bowsprit end, as if it is longer than that distance the excess is taxed in computing the area of the fore triangle for head sail under the Y.R.A. rules.

Length of Spars for Yawls and Ketches.

The mainmast of a yawl in length, deck to hounds, is about 2·9 times her beam; although racing yawls have sometimes a greater proportionate length of mast than cutters, the proportion being simply as $\sqrt{\text{mainsail area}}$.

The length of main boom is governed by the distance between the mainmast and mizenmast. The common proportion is $\sqrt{\text{mainsail area}} \times 1{\cdot}1$.

Main gaff about ·75 of the length of main boom. The length of the mizenmast of a yawl, from deck to hounds, is usually ·7 of the length of the mainmast deck to hounds; occasionally, however, it is no more than ·6 of the length of mainmast. The mizen is ·5 of the length of main boom. Mizen yard ·65 of the length of main gaff; or, if a gaff is used, as is now generally the case, ·5 the length of main gaff.

The bowsprit of a yawl is generally about ·4 of the length on the load water-line, and rarely is as long as the bowsprit of a cutter of similar size. The first impression would be that a yawl, owing to her mizen being stepped so far aft, would require a longer bowsprit than a cutter, but in practice this is not found to be the case. In fact, as a rule, a yawl carries a very slack helm on a wind in moderate breezes, and the reason is that the mast is usually stepped a little farther forward than in a cutter; and, besides this, the eddy wind from the mainsail strikes the fore cloths of the mizen sail from to leeward, so that the luff of the sail lifts, and this unsteadiness in turn disturbs the currents of air that are approaching the sail from to windward, the final result being that for close-hauled sailing the mizen is a much less effective sail than is generally supposed.

The mainmast of a ketch in length, deck to hounds, is about 2·7 times her beam, or equal to $\sqrt{\text{mainmast area}}$, simply. The main boom is usually the length of the distance between the masts and the gaff, from ·8 to ·9 of the length of main boom. The mizenmast, deck to hounds, from ·8 to ·85 of the mainmast. The mizen beam is from ·75 to ·85 of the length of mainboom; and mizen gaff ·75 to ·85 of the length of mizen boom.

The ketch rig is not often adopted for yachts, one reason probably

* Called a working topsail because a vessel can work to windward with it in a whole sail breeze.

being that, the mizenmast being stepped forward of the sternpost, it comes very much in the way of the tiller, and the mizen boom has to be kept a great height above the deck. Another objection to the ketch rig is that the sails are necessarily narrow in proportion to their height, and for this reason their heeling moment for any given area is greater than is the heeling moment of the squarer sails of a cutter or yawl. The ketch rig also compares badly with the yawl rig for plying to windward, especially in a head sea, the long gaff and smallness of the area of the mainsail compared with the total sail area probably having something to do with this. On the other hand, a ketch is handier under head sail and mizen than a yawl; and, the sails being more equally divided, she can be worked with a smaller number of men or any given total sail area.

Spars for cruisers are usually from 8 to 10 per cent. smaller than those of racers.

In determining upon the length of main boom, the designer of the sail plan will be guided by the figure of the sails; nevertheless, it must be always borne in mind that the shorter the boom is, the lighter it will be, and the more the weights—which are necessarily above the centre of gravity—can be reduced, the stiffer the vessel will be. Some confusion, however, occasionally appears to exist about this matter in consequence of the erroneous supposition that the quarters of a vessel bear the weight of the boom by some independent support, and sailors frequently say of a vessel, "She'll never bear the weight of that great boom with those lean quarters." It has already been pointed out that any weight a vessel carries has its support through the common centre of gravity of the vessel (Fig. 12, page 23); and lean quarters will no more interfere with a vessel carrying a long and heavy main boom than a lean bow will. But a long, heavy boom, the same as a long, heavy bowsprit, will tend to increase the momentum acquired during pitching and scending in a sea way; and, moreover, it will also tend to increase this momentum because a main boom is somewhat free in its action, and is not stayed in an immovable position, the same as a mast or a bowsprit is. Then, again, a main boom is always more or less off the lee quarter, and thus assists in heeling a vessel; and, of course, the heavier a boom is, the more potent this assistance will be. But a "leanness" of the quarters has nothing to do with this matter, further than that lean quarters affect the general question of stability.

In giving a vessel a light boom, however, it should never be lost sight of that about the worst thing in a vessel's outfit is a weak boom. With a weak boom the mainsail can never be made to stand properly if there be the least semblance of a breeze; and, moreover, so far as danger to the

crew goes, a worse accident cannot happen to the spars than the breaking of the main boom. The modern practice, however, of lacing the sail to the boom somewhat diminishes the danger of a broken boom.

Girths of Spars.

The diameter or girths of spars vary a great deal according to the pine chosen, but the general fashion now is, except in yachts of 52ft. rating and under, to have Oregon or Vancouver for masts, and the same for booms and bowsprits. In smaller vessels white Baltic pine is used for masts, booms, and bowsprits. The diameter of schooners' masts at the deck is generally from ·023 to ·025 of the length from deck to hounds; thus, say a schooner has a mast 30ft. from deck to hounds, then the diameter at the deck will be $30 \times \cdot 025 = 0{\cdot}75$ft. or 9in.; this will be for tough Oregon or Riga pine; for pines of less breaking strength ·027 will be a better proportion. The diameter of the mast at the hounds is generally about ·85 of the diameter at the deck; or, if the diameter at the deck be ·75ft., then $\cdot 75 \times \cdot 85 = \cdot 637$ft. or 7⅝in.

The diameter of a cutter's mast at the deck varies from ·022 of the length deck to hounds for 36ft. raters, of the length deck to hounds, and from ·028 to ·031 for yachts of 80ft. rating and upwards. In cutters the diameter at the hounds is usually ·9 that at the deck, but in racing yachts the diameter is generally equal all the way up. For instance, in the Britannia the diameter at the deck is 20 inches, and at the hounds 22¾ inches; the mast, however, is slightly oval at the hounds, but only to the extent of an inch or so below the hounds. The diameter at the upper cap is usually ·8 of that at the hounds.

The diameter of a yawl's mast is from ·022 in small vessels to ·028 in large of the length deck to hounds.

The diameter of topmasts at the heel is ·02 the length heel to hounds and the diameter at the hounds is ·6 or ·7 of the diameter at the heel.

The bowsprit is more tapering, and the diameter at the stem head varies from ·028 to ·04 of the length of bowsprit outside, whilst the diameter at the sheave or outer end ranges from ·7 to ·8 of the diameter at the stem head.

The diameter of gaffs varies from ·016 of the length in small vessels to ·02 in large. The diameter at the outer end is ·7 to ·8 of the inner end.

The diameters of main booms vary considerably, and of course will be greatest if the sails be not laced. A cutter's main boom generally has a greatest diameter near the sheet, or ¾ the length from the inner end, of ·016 to ·02 of its length, if the sail is not laced; the diameter of the fore end, next the mast, is usually about ·75 of the greatest diameter; whilst

the diameter of the outer end is also about ·75 of the greatest diameter. The boom really tapers very slowly from the sheet towards the inner end until about one-seventh its length from the mast when it tapers faster; and the same for the outer end. A yawl's main boom in diameter is usually ·016 of its length. For a laced sail the diameter of a boom is generally reduced one-fifth.

The diameter of a topsail yard at its centre of a length is ·015 its whole length. The diameter at the ends ·73 the greatest diameter.

A balloon-topsail yard, or the big working-topsail yard, is generally strengthened amidships by battens of American elm, put on with seizings.

The diameter of a spinnaker boom is generally from ·012 to ·014 of its length, and the diameter at the ends ·85 of the greatest diameter.

The general plan of constructing spars and their fittings will be found on Plates XXX., XXXI., XXXII., &c.

A great many contrivances have at various times been introduced with a view of lightening spars; but the old-fashioned "grown stick" has not yet been displaced. The Black Maria, in 1852, appeared with "built" spars, something on the principle of a cooper's cask; but the plan gained no favour among English yachtsmen. Later—we think it was in 1863—hollow steel spars were tried in England; but, after two or three masts were carried away rather suddenly, funnel masts were unceremoniously discarded as much too dangerous. In 1868 some one proposed the boring of masts, and Mr. Michael Ratsey was commissioned to bore the masts and bowsprit of the Cambria. The boring was performed by long augers inserted at either end, and meeting in the centre of the spar's length. The "bore" was about 4in. in diameter, and no doubt the spars were very much reduced in weight thereby. The Egeria afterwards had her masts bored; but the plan was speedily condemned, as in the following year the Cambria's foremast-head tumbled off. However, this did not deter the owner of the Cambria trying bored spars again; but Mr. Ratsey thought it prudent to put three heavy iron bands round the mastheads at equal intervals, and to have masthead pendants, in addition to the usual pendants and runners. This had the effect of strengthening the mastheads; but the weight of material used in so strengthening them was equal in weight to the centre piece which had been abstracted from the spars; and, moreover, the greater portion of the weight was concentrated at the masthead, instead of being distributed throughout the whole length of the spar. Yachtsmen thereupon not unnaturally concluded that no benefit was to be derived from bored spars; but the Cambria continued to carry them, and crossed the Atlantic twice with them. In 1876 the John Harvey Company, of Wivenhoe, introduced a new kind

of built spar; the spar was sawn through its whole length, and a piece taken out of each half, tapering to correspond with the tapering of the spar. All kind of "built" mainbooms have since been used in racing yachts, including lately some of steel, but a good Oregon stick is still mostly in favour.

PLACING THE MASTS.

With regard to the placing of the masts, a great many rules have been given; but the soundest advice is to keep them as near to the fore and aft position of the centre of gravity of the vessel as possible, if it is sought to make a yacht easy in a sea-way. The weight of the masts, by lengthening out the radius of gyration, much increases the violence of pitching; and the shorter the masts are, and the nearer they are stepped to the centre of gravity of the vessel, the less they will aggravate pitching. A quarter of a century ago, the America introduced the fashion of raking masts aft, and no doubt there was some reason in it, if it were absolutely necessary that masts should be stepped very far forward. By raking the masts aft, their weight was carried aft in proportion to the sine of the angle of rake, and so, although a mast might have been stepped very far into the bows of a vessel, yet would its weight fall very near the centre of gravity. However, experience soon undid this fashion, as it was ascertained that sails of better figures, which could be more easily worked, were to be obtained by stepping the masts a trifle farther aft, and upright.

So far as cutters are concerned, the general rule appears to be to step the mast 0·4 of the length on the load water-line abaft the fore side of the stem, with a rake aft. from 1° to 2°. This position is found to be near enough to the centre of gravity of a vessel to avoid any bad effect on her ease in a seaway; and generally, if any good results through stepping a mast 6in. further forward or aft, the good is traceable to the altered position of the centre of effort of the sails rather than to the shift that has been made to the weight of the mast. Of course, if the centre of effort of the sails cannot be altered or adjusted, without shifting the mast, this extreme measure must be adopted; but the constant craze some sailing masters have for shifting masts backwards and forwards should never be gratified from the mere assertion of "I'm sure the vessel would do a great deal better if her mast was moved six inches further aft; we shifted the Anonyma's six inches aft, and she went as well again afterwards." This is usually the skipper's "theory," and, as the builder is very properly anxious to do anything to improve his construction, the mast is generally shifted after the skipper's sententious opinion. A mast may require shifting; but often a little alteration to the fore foot, or a trifling alteration

to the sail plan, will correct a vessel's indifferent performance and handiness when sailing on a wind.

The mast of a yawl is generally placed a little farther forward, or about ·38 of the length on the load line abaft the fore side of the stem. The mast is placed so far forward in order that the mainsail may be as large as the Yacht Racing Association Rules permit. The mizen mast of a yawl generally has a slight rake aft, in order that the main boom may clear it, as frequently the length of counter will not admit of it being stepped sufficiently far aft to clear the boom if no rake be given.

With regard to schooners, the present plan, in order to secure good weatherly qualities, is to make them as much like cutters as possible. In racing schooners at least the comfort of a good large boom foresail is an unknown thing; but the advantage of having a big mainsail, or as much canvas as possible in one piece, is too great to be sacrificed for comfort. In a like way the exigencies of yacht racing having gradually turned the once comfortable yawl, with her mizen stepped close to her rudder-head, into a vessel that, so far as her ease in a sea is dependent on her spars, might better be a cutter.

The masts of a ketch are thus placed by the Y.R.A. rules; mainmast at the centre of length of load water-line; mizen-mast 0·4 abaft the centre of length. This rule brings the masts of a ketch half of the length of the load line apart, and the Y.R.A. rule was founded upon it.

The sizes given for blocks in the following table are for cutters; for yawls of equal tonnage take the sizes set down for the tonnage next below; thus, say a yawl is 100 tons, then take the sizes for blocks from the column assigned to 80 tons. (The general arrangement of rigging is shown on Plate XXXVII.).

TOPPING LIFTS.

In yachts above 50 tons there is a block on the hauling end of the topping lift, through which a runner is rove. One end of this runner is shackled to an eye-bolt in the channel; at the other end there is a purchase tackle, the lower block of which is also shackled to an eye-bolt in the channel. The fall of the purchase is belayed on a pin in the cavel inside the rigging. In large vessels of 100 tons and upwards there is a block locked to an eye-bolt in main-boom end, and on cheeks of mast (each side). One end of the topping lift is shackled to the block on the cheek of the mast; then rove through the block on the boom end, up through the block on the cheek of mast. At the end a block is shackled with the runner and tackle, as in the other case. (*See* Table III.). For details of the other rigging the reader is referred to "Yacht and Boat Sailing," last edition.

TABLE I.—BLOCKS FOR RACING YACHTS BY Y.R.A. RATING.

	Description of Blocks.	No. of Single Blocks.	No. of Double Blocks.	18ft. Rating.	24ft. Rating.	30ft. Rating.	36ft. Rating.	42ft. Rating.	52ft. Rating.	65ft. Rating.	60 Rating.	80 Rating.	100 Rating.	120 Rating.
				Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.
Throat halyards	Shackle	—	2*a*	2	2¾	4½	5	5½	7	8	9	10	10½	11
Throat purchase	Shackle	1	1	2	2¾	3	3½	4	5	6	6½	7	7½	8
Peak halyards	Hook	5	—	—	—	4½	5	5½	7	8	8½	9½	10½	10½
Peak purchase	Shackle	1	1	—	—	3	3½	4	5	6	6½	7	7½	8
Fore halyards	Cliphook	2*b*	—	2	2½	3	3½	4	5	6	6½	7	7½	8
Jib halyards	Shackle	3*b*	—	—	—	4	4½	4½	6	7½	8½	9	9½	10
Jib purchase	Shackle	1	1	—	—	3	3½	4	5	6	6½	7	7½	8
Runner pendant	Lugs	2	—	2	2½	3	3½	4	5	6	6½	7	7½	8
Runner tackles	Hook & Fiddle	2	2*c*	—	2½	3 & 7	3½ & 7½	4 & 7½	4½ & 8	5 & 9	5½ & 10	6 & 11	6½ & 12	7 & 13
Topping lift	Hook	2	—	2	2½	4	4½	5	5½	6	6½	7	7½	8
Topping lift purchase	Lugs	2	2	—	—	3	3½	4	5	5½	6	6½	7	7½
Main sheet	Lug and strop	—	2*d*	2½	3	4½	5	5½	7	8½	9½	10½	11	12
Main sheet leads	Swivel	2	—	—	2½	3	3½	4	5	6	7	7½	8	8½
Topmast backstays	Shackle	4	4	—	—	3	3½	4	5	5½	6	6½	7	7½
Preventer backstays	Shackle	4	—	—	—	3	3½	4	5	5½	6	6½	7	7½
Topsail sheet	Shackle	1	—	—	—	3	3½	4	5	5½	6	6½	7	7½
Topsail halyards	Hook	1	—	—	—	3½	4	4½	5	5½	6	6½	7	7½
Fore sheet	Hook	2	—	2	2	3	3½	4	4½	5	5½	6	6½	7
Reef tackle	Hook & Fiddle	1	1	2	2¾	3 & 7	3½ & 7½	4 & 7½	4½ & 8	5 & 9	5½ & 10	6 & 11	6½ & 12	7 & 13
Jib tack	Hook	1	1	2	2¾	3	3½	4	5	5½	6	6½	7	7
Main outhaul	Hook	1	1	2	2¾	3	3½	4	5	5½	6	6½	7	7
Trysail sheets	Hook	2	2	2	3	4	4½	5	6	7	7½	8	8½	9
Bobstay	Steel shackle	1	1	2	2¾	4	4½	5	5½	6	6½	7	7½	8
Bowsprit shrouds	Shackle & Fiddle	2	2	2	2¾	3 & 7	3½ & 7½	4 & 7½	4½ & 8	5 & 9	5½ & 10	6 & 11	6½ & 12	7 & 13
Sundry tackles	Rope strop	12	12	2	2¾	3	3½	4	5	5½	6	6½	7	7½
Gaff bulls' eyes	Iron bound	2	—	—		3	3	3½	4½	5	5½	6	6½	7
Jib sheet ditto	Iron bound	2	—	—	—	3	3	3	4	4½	5	5½	6	6½
Spinnaker halyards, spinnaker guys, and jib topsail halyards and down-hauls	Rope strop	7	—	2	2½	3	3	3½	4	4½	5	5½	6	6½

(*a*) In yachts of 42ft rating and over where a throat purchase is used, the upper block has three sheaves, as there is no standing part to the halyards. The blocks, if of steel, are about 20 per cent. smaller in the shell, but the sheaves are full size. In yachts of 36ft. rating down to 1 rating the halyards are of flexible wire and the blocks of steel. (*b*) These are steel blocks. (*c*) The double block is fiddle-shaped. (*d*) The upper block has three sheaves, and wire rope stropped. The lower block has lugs.

TABLE II.—SIZES FOR STANDING AND RUNNING RIGGING FOR RACING YACHTS.

	Description.	18ft. Rating.	24ft. Rating.	30ft. Rating.	36ft. Rating.	42ft. Rating.	52ft. Rating.	65ft Rating.	80ft. Rating.	90ft. Rating.	100 t. Rating.	120ft. Rating.
		Circumference. Inches.	Circumference. Inches.	Circumference. Inches.	Circumference. Inches.	Circumference. Inches.	Circumference. Inches.	Circumference. Inches.	Circumference. Inches.	Circumference. Inches.	Circumference. Inches.	Circumference. Inches.
Shrouds *a*	Crucible steel	1/2	3/4	1 1/4	1 1/4	1 3/8	1 1/2	1 3/4	2	2 1/4	2 1/2	3
Runner pendants	Crucible steel	3/8	1/2	1 1/8	1 1/4	1 3/8	1 1/2	1 3/4	2	2 1/4	2 1/2	3
Forestay	Crucible steel	3/8	1/2	1 1/8	1 1/8	1 1/4	1 3/4	1 3/4*c*	2*c*	2 1/4*c*	2 1/2*c*	2 3/4*c*
Bowsprit shrouds	Crucible steel	1/2	3/4	1 1/8	1 1/4	1 3/8	1 5/8	1 3/4	2	2 1/4	2 1/2	2 3/4
Bobstay pendant	Crucible steel	1/2	3/4	1 1/4	1 1/8	1 3/4	2	2 1/4	2 1/2	2 3/4	3	3 1/4
Topmast backstays	Crucible steel	—	—	5/8	3/4	1	1 1/8	1 1/4	1 3/8	1 1/2	1 5/8	1 3/4
Preventer backstays	Crucible steel	—	—	5/8	3/4	7/8	1	1 1/8	1 3/8	1 1/8	1 5/8	1 3/4
Topmast stay	Crucible steel	—	—	3/4	7/8	1	1 1/8	1 1/4	1 3/8	1 1/8	1 5/8	1 3/4
Topping lifts	Flexible steel	3/8	1/2	5/8	3/4	7/8	1	1 1/4	1 5/8	1 1/8	1 5/8	1 3/4
Jib halyards	Flexible steel	—	—	7/8	1	1 1/8	1 1/4	1 1/8	1 3/4	1 7/8	2	2 1/4
Topsail halyards	Flexible steel	—	—	7/8	1	1 1/8	1 1/4	1 1/8	1 3/4	1 7/8	2	2 1/4
Runners	Flexible steel	—	—	7/8	1	1 1/8	1 1/4	1 1/8	1 3/4	1 7/8	2	2 1/4
Bobstay tackle	Flexible steel	—	—	7/8	1	1 1/4	1 1/8	1 3/4	2	2 1/4	2 3/8	2 1/4
Jib tack	Flexible steel	3/8	1/2	7/8	1	1 1/4	1 1/8	1 3/4	2	2 1/4	2 3/8	2 1/4
Throat halyards	Hemp*b*	1/2	3/4	1 1/4	1 1/8	2	2 1/8	2 3/4	3 1/4	3 1/8	3 3/4	4
Peak halyards	Hemp	3/4	3/4	1/2	3/4	2	2 1/8	2 1/8	3	3 1/4	3 1/8	4
Fore halyards *b*	Hemp	3/4	1/2	1/2	3/4	1 1/8	1 3/4	2 1/4	2 1/8	2 3/4	3	3 1/4
Fore sheets	Hemp	3/4	1	1	1 1/4	1 1/8	2	2 1/4	2 1/8	2 3/4	3	3 1/4
Jib sheets	Hemp	—	—	1	1 1/8	1 3/4	2 1/4	2 1/8	2 3/4	3	3 1/4	3 1/8
Purchases	Hemp	3/4	1	1	1	1 1/4	1 1/2	1 3/4	2	2 1/4	2 1/2	2 3/4
Spinnaker and jib topsail gear	Hemp	3/4	1	1	1	1 1/4	1 1/8	1 3/4	2	2 1/4	2 1/2	2 3/4
Spinnaker guys	Hemp	3/4	1	1 1/4	1 1/4	1 1/2	1 3/4	2	2 1/4	2 1/8	2 3/4	3
Main sheet	Manilla	3/4	1	1 1/8	1 3/4	2	2	2 1/8	2 3/4	3	3 1/4	3 1/4
Runner tackles	Manilla	—	—	1 1/8	1 1/8	1 3/4	2	2 1/4	2 1/8	2 3/4	3	3 1/4
Topsail sheet	Manilla	—	—	1 1/8	1 3/4	1 3/4	2	2 1/4	2 1/8	2 3/4	3	3 1/4
Topping lift tackles	Manilla	3/4	1	1 1/4	1 1/4	1 1/8	2	2 1/4	2 1/8	2 3/4	3	3 1/4
Diameter of rigging screws	Gn. mtl. & steel	3/8	1/2	1/2	5/8	3/4	7/8	1 1/8	1 1/4	1 3/8	1 1/16	1 1/8
Ditto topmast backstays	Gn. mtl. & steel	—	—	5/16	3/8	1/2	5/8	3/4	7/8	1	1 1/16	1 1/8

(*a*) One on each side only in 0·5-rater and 1-rater; two 2·5-rater and 5-rater. (*b*) Flexible steel in 1 to 5-rating. Hemp or manilla in 18ft. rating. Manilla can in all cases be substituted for hemp if preferred. 36ft. raters (see Plate XXXI., &c.) a double whip of 2in. manilla and 1 3/8in. manilla purchase is used. (*c*) Double forestay.

In using these tables, Table VII. should be consulted.

TABLE III.—SIZES OF BLOCKS FOR YACHTS OF VARIOUS SIZES.

Name of Block.	Description. Number. Double.	Description. Number. Single.	Tons. Cut. 3. Yawl 5. Size.	Tons. Cut. 5. Yawl 7. Size.	Tons. Cut. 10. Yawl 14. Size.	Tons. Cut. 15. Yawl 20. Size.	Tons. Cut. 20. Yawl 30. Size.	Tons. Cut. 30. Yawl 40. Size.	Tons. Cut. 40. Yawl 60. Size.	Tons. Cut. 60. Yawl 80. Size.	Tons. Cut. 80. Ywl. 100. Size.	Tons. Cut. 100. Ywl. 130. Size.
			IRON STROP BLOCKS.									
			Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
Throat halyards *a*	2	—	4	4½	5	5½	6	7	8	9	9½	10
Peak halyards	—	5	4	4½	5	5½	6	7	8	9	9½	10
Main sheet	1	1	4½	4½	5	5½	6	7	8	9	9½	10
Main sheet lead	—	2	3	3	4	4½	5	5½	6	6½	7	7½
Jib halyards	—	3	4	4½	5	5½	6	7	8	9	9½	10
Fore halyards	—	2	3	3	4	4½	5	6	6½	7	7½	8
Bobstay (iron blocks)	1	1	4	4½	5	5½	6	7	8	9	9½	10
Bowsprit shrouds *b*	2	2	6 & 3	6 & 3	6½ & 4	7 & 4½	7½ & 5	8 & 5½	8½ & 6	9 & 6½	9½ & 7	10 & 7½
Pendant blocks	—	2	3½	3½	4	4½	5	5½	6	6½	7	7½
Runner tackle *b*	2	2	6 & 3	6 & 3	6½ & 4	7 & 4½	7½ & 5	8 & 5½	8½ & 5½	9 & 6	9½ & 6½	10 & 7
Main outhaul	—	1	3	3	3½	4	4½	5	5½	6	6½	7
Topsail sheet (cheek)	—	1	3	3	3½	4	4½	5	5½	6	6½	7
Topmast backstays *c*	4	4	3	3	3½	4	4½	5	5½	6	6½	7
Preventer backstays	2	2	3	3	3½	4	4½	5	5½	6	6½	7
Preventer backstay whips	—	2	3	3	3½	4	4½	5	5½	6	6½	7
Jib tack	—	1	3	3	3½	3½	4	4½	5	5½	6	6½
Jib purchase	1	1	3	3¼	3½	4	4½	5	5½	6	6½	7
Jib purchase runner	—	1	—	—	—	—	—	—	5	6	7	8
Main purchase	1	1	3	3¼	3½	4	4½	5	5½	6	6½	7
Peak purchase	1	1	3	3¼	3½	4	4½	5	5½	6	6½	7
Topping lifts	—	2	3	3¼	—	4	4½	5	5½	6	6½	7
Topping lift purchase *d*	4	—	3	3	3½	4	4½	5	5½	6	6½	7
Spinnaker topping lift *e*	2	—	3	3	3½	4	4½	5	5½	6	6½	7
			ROPE STROP BLOCKS.									
Jib topsail halyards	—	2	3	3	3½	4	4½	5	5½	6	6½	7
Spinnaker halyards	—	1	3	3	3½	4	4½	5	5½	6	6½	7
Spinnaker guy whips	—	2	3	3	3	3½	4	4½	5	5½	6	6½
Trysail sheets	2	2	4	4½	5	5½	6	6½	7	7½	8	8½
Fore sheets	—	4	4	3½	4	5	5½	6	6½	7	7½	8
Reef-tackle (fiddle dbl.)	1	1	6 & 3	6 & 3½	6½ & 4	7	7½	8	8½	9	9½	10
Boom guy	1	1	—	—	3	3½	4	4½	5½	6½	7	7½
Tack tackles	3	3	3	3	3½	4	4½	5	5½	6	6½	7
Burton	—	2	—	—	3	4	4½	5	5½	6	6½	7
Gaff topsail sheet whip	—	1	3	3	3½	4	4½	5	5½	6	6½	7
Downhauls (forward)	—	3	—	—	3	4	4½	5	5½	6	6½	7
Dead-eyes *f*	—	—	3¾	3¾	4	4¼	4½	5	5¾	6½	7½	8½

(*a*) If a throat purchase is used (as it generally is in racing yachts of 10 tons and over, the upper throat halyard block is a threefold, as one part of the halyards is required for the purchase, and another for the hauling part.

(*b*) The double blocks on the bowsprit shrouds and runner tackle are fiddles.

(*c*) A five-tonner has only four single blocks for backstays, two on each side.

(*d*) These would be single up to 20 tons.

(*e*) These would be single up to 10 tons.

(*f*) Some five-tonners have only two shrouds on each side; 10 tons and upwards three shrouds; above 60 tons four shrouds for racing.

The sizes of cordage are arranged suitably for Thames tonnage, but Table VII., page 517, should be consulted before deciding upon the cordage.

TABLE IV.—SIZES OF CORDAGE FOR CUTTER AND YAWL YACHTS OF VARIOUS SIZES BY THAMES TONNAGE.

NAME OF ROPE.	TONS. Cut. 3. Yawl 5.	TONS. Cut. 5. Yawl 7.	TONS. Cut. 10. Yawl 14.	TONS. Cut. 15. Yawl 20.	TONS. Cut. 20. Yawl 30.	TONS. Cut. 30. Yawl 40	TONS. Cut. 40. Yawl 60.	TONS. Cut. 60. Yawl 80.	TONS. Cut. 80. Ywl. 100	TONS. Cut. 100. Ywl. 130
	Cf.	Cf.	Cf.	Cf.	Cf.	Cf.	Cf.	Cf.	Cf.	Cf.
	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
Throat halyards	1½	1¾	2	2¼	2½	2¾	3	3¼	3½	3¾
Peak halyards	1½	1¾	2	2¼	2½	2¾	3	3¼	3½	3¾
Main sheet (manilla)	1½	1½	1¾	2	2¼	2½	2¾	3	3¼	3½
Fore halyards	1	1¼	1½	1¾	1¾	2	2¼	2½	2¾	3
Bobstay tackle †	1½	1¾	2¼	2½	2¾	3	3¼	3½	3¾	4
Bowsprit shrouds tackle	1	1¼	1¼	1½	1¾	2	2¼	2½	2¾	3
Pendant	1½	1¾	2	2¼	2½	2¾	3	3¼	3½	3¾
Runner	1½	1¾	2	2½	2¾	3	3¼	3½	3¾	4
Runner tackle	1	1¼	1½	1¾	1¾	2	2¼	2½	2¾	3
Main outhaul	1	1	1¼	1½	1½	1¾	2	2¼	2½	2¾
Reef pendants	1¾	1¾	2	2¼	2½	2½	3	3¼	3½	3¾
Topsail sheet	1	1¼	1½	1¾	2	2¼	2½	2¾	3	3¼
Topmast backstay tackles	1	1¼	1¼	1½	1½	1¾	2	2½	2¾	3
Preventer backstay tackles	1	1¼	1¼	1½	1½	1¾	2	2½	2¾	3
Preventer backstay whips	1	—	1¼	1½	1½	1¾	2	2½	2¾	3
Jib tack	1¾	1¾	2	2¼	1¼ *w*	1½ *w*	1¾ *w*	2 *w*	2¼ *w*	2½ *w*
Jib halyards (chain)	3/16	3/16	3/16	¼	5/16	⅜	⅜	7/16	½	½
Jib halyards (manilla)	1¾	1¾	2	2¼	2½	2¾	3	3¼	3½	3¾
Jib sheets	1½	1½	1¾	2	2¼	2½	2[illegible]	3	3¼	3½
Jib purchase	1	1¼	1¼	1½	1½	1¾	1¾	2	2¼	2½
Jib purchase runner	—	—	—	—	—	—	1½ *w*	1¾ *w*	2 *w*	2¼ *w*
Throat purchase	1	1¼	1¼	1½	1½	1¾	1¾	2	2¼	2½
Peak purchase	1	1¼	1¼	1½	1½	1¾	1¾	2	2¼	2½
Topping lifts	1½	1¾	2	2½	2¾	3	3¼	3½	3¾	4
Topping lift purchase	1	1¼	1¼	1½	1½	1¾	1¾	2	2¼	2½
Jib topsail halyards	1	1¼	1¼	1½	1½	1¾	1¾	2	2¼	2½
Spinnaker halyards	1	1¼	1¼	1½	1½	1¾	1¾	2	2¼	2¼
Spinnaker guy whips	1	1¼	1¼	1½	1¾	2	2½	2¾	2¾	2¾
Spinnaker topping lift	1	1¼	1¼	1½	1½	1¾	1¾	2	2¼	2½
Trysail sheets	1	1¼	1¼	1½	1¾	2	2¼	2¾	3	3¼
Fore sheets	1	1¼	1¼	1½	1[illegible]	2	2¼	2¾	3	3¼
Reef tackle (fiddle)	1	1	1¼	1½	1½	1¾	1¾	2	2¼	2½
Boom guy	1	1	1¼	1½	1½	1¾	1¾	2	2¼	2½
Tack tackles	1	1	1¼	1½	1½	1½	1¾	1¾	2	2¼
Burton	—	—	—	1½	1½	1¾	1¾	2	2¼	2½
Gaff topsail sheet whip	1½	1¾	2	2¼	2½	2¾	3	3¼	3½	3¾
Downhauls, peak, fore sail, and jib	1	1	1	1	1¼	1½	1½	1¾	2	2
Lanyards	1	1	1¼	1½	1¾	2	2¼	2½	2¾	3
Topsail halyards	1¼	1¼	1½	1¾	1¾	2	2¼	2½	2¾	3
Topsail trip halyards	1	1	1¼	1½	1½	1¾	1¾	2	2¼	2½

(*w*) Flexible steel wire jib tack and jib purchase runner. Cf. Circumference.

In the case of jib halyards the size of the iron of the link is given.

† Or equivalent in flexible steel wire.

TABLE V.—CIRCUMFERENCE IN INCHES OF STEEL WIRE FOR STANDING RIGGING FOR CUTTERS AND YAWLS.

NAME.	TONS. Cut. 3 or 5 Yawl 7.	TONS. Cut. 10. Yawl 14.	TONS. Cut. 15. Yawl 20.	TONS. Cut. 20. Yawl 30.	TONS. Cut. 30. Yawl 40.	TONS. Cut. 40. Yawl 60.	TONS. Cut. 60. Yawl 80.	TONS. Cut. 80. Ywl. 100	TONS. Cut. 100. Ywl. 120	TONS. Cut. 120 Ywl 140
Shrouds	1¼	1⅜	1½	1¾	2¼	2½	3	3	3½	3¾
Pendants	1¼	1⅜	1¾	2	2¼	2½	3¼	3½	3¾	4
Bowsprit shrouds	1¼	1⅜	1¾	2	2¼	2½	2¾	3	3¼	3½
Forestay	1½	1¾	2	2¼	2½	2¾	3¼	3½	3¾	4
Bobstay pendant	1½	1¾	2	2¼	2½	2¾	3¼	3½	3¾	4
Topmast stay	¾	¾	1	1	1¼	1½	1¾	2	2¼	2½
Topmast backstay	¾	¾	¾	1	1¼	1½	1¾	2	2¼	2½
Preventer backstay ...	¾	¾	1	1	1¼	1½	1¾	2	2¼	2½
No. of shrouds a side...	2	2	3	3	3	3	3	4	4	4
No. of backstays a side	1	1	2	2	2	2	2	2	2	2
Copper bobstay bar ...	⅝	¾	⅞	1	1¼	1⅜	1½	1¾	2	2¼

TABLE VI.—RELATIVE SIZE AND STRENGTH OF HEMP, IRON, AND STEEL ROPE.

Hemp Rope.		Iron Wire Rope.		Steel Wire Rope.		Equivalent Strength.	
Circumference.	Pounds weight per Fathom.	Circumference.	Pounds weight per Fathom.	Circumference.	Pounds weight per Fathom.	Working Load in Cwts.	Breaking strain in Tons.
2¾	2	1	1	...	...	6	2
...	...	1¼	1¼	1	1	9	3
3¾	4	1⅝	2	...	...	12	4
...	...	1¾	2½	1¼	1¼	15	5
4½	5	1⅞	3	...	...	18	6
...	...	2	3½	1⅜	2	21	7
5½	7	2⅛	4	1½	2½	24	8
...	...	2¼	4½	...	...	27	9
6	9	2⅜	5	1⅞	3	30	10
...	...	2½	5½	...	...	33	11
6½	10	2⅝	6	2	3½	36	12
...	...	2¾	6½	2⅛	4	39	13
7	12	2⅞	7	2¼	4½	42	14
...	...	3	7½	...	...	45	15
7½	14	3⅛	8	2⅜	5	48	16
...	...	3¼	8½	...	...	51	17
8	16	3⅜	9	2½	5½	54	18
...	..	3½	10	2⅝	6	60	20
8½	18	3⅝	11	2¾	6½	66	22
...	...	3¾	12	...	.	72	24
9½	22	3⅞	13	3¼	8	78	26
10	25	4	14	...	...	84	28
...	...	4¼	15	3⅜	9	90	30
11	30	4⅜	16	..	...	96	32
...	...	4½	18	3½	10	108	36
12	36	4⅝	20	3¾	12	120	40

TABLE VII.—DIMENSIONS OF SPARS TO WHICH THE BLOCKS AND CORDAGE GIVEN IN THE PRECEDING TABLES ARE SUITABLE.

NAME.	TONS. Cut. 3. Yawl 5.	TONS. Cut. 5. Yawl 7.	TONS. Cut. 10. Yawl 14.	TONS. Cut. 15. Yawl 20.	TONS. Cut. 20. Yawl 30.	TONS. Cut 30. Yawl 40.	TONS Cut. 40. Yawl 60.	TONS. Cut. 60. Yawl 80.	TONS. Cut. 80. Ywl. 100	TONS. Cut. 100. Ywl. 120
	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.
Mast, deck to hounds...	21	23	27	30	34	37	40	44	47	50
Topmast, fid to hounds.	17	19	24	28	29	30	33	36	40	44
Main boom	27	29	33	38	43	47	51	55	61	68
Main gaff	18	20	22	25	28	30	33	36	40	45
Bowsprit outboard......	16	17	18	21	24	25	27	30	32	34
Topsail yards	{ 27	30	35	36	37	38	39	41	43	46
	{ 18	20	21	22	23	24	25	26	28	30
Spinnaker boom.........	30	35	40	42	44	45	46	50	54	58

TABLE VIII.—SCHOONERS.

CIRCUMFERENCE IN INCHES OF STEEL WIRE FOR STANDING RIGGING.

R. Racer. C. Cruiser.	TONS. R. 20. C. 30.	TONS. R. 30. C. 40.	TONS. R. 40. C. 60.	TONS. R. 50. C. 75.	TONS. R. 75. C. 100.	TONS. R. 100. C. 140.	TONS. R. 125. C. 180.	TONS. R. 150. C. 220.	TONS. R. 175. C. 250.	TONS. R. 200. C. 275.
Shrouds	1½	1½	1¾	2¼	2½	3	3	3½	3¾	4
Pendants	1½	1¾	2	2¼	2½	3¼	3½	3¾	4	4¼
Bowsprit shrouds	1½	1¾	2	2¼	2½	2¾	3	3¼	3½	3¾
Forestay	1¾	2	2¼	2½	2¾	3¼	3½	3¾	4	4¼
Bobstay pendant	1¾	2	2¼	2½	2¾	3¼	3½	3¾	4	4¼
Topmast stay	¾	1	1	1¼	1½	1¾	2	2¼	2½	2¾
Topmast backstay	¾	¾	1	1¼	1½	1¾	2	2¼	2½	2¾
Topmast preventer backstay	¾	1	1	1¼	1½	1¾	2	2¼	2½	2¾
No. of shrouds a side.........	2	3	3	3	3	3	3	3	3	3
No. of backstays a side......	1	2	2	2	2	2	2	2	2	2
Copper bobstay bar	¾	⅞	1	1¼	1⅜	1½	1¾	2	2¼	2½

Table IX.—SCHOONERS.

MAINMAST.

R. Racer. C. Cruiser.	Description.		Tons R. 20. C. 30.	Tons R. 30. C. 40.	Tons R. 40. C. 60.	Tons R. 50. C. 75.	Tons R. 75. C. 100.	Tons R. 100. C. 140.	Tons R. 125. C. 180.	Tons R. 150. C. 220.	Tons R. 175 C. 250	Tons R. 200 C. 275
Name of Block.	Number. Double.	Number. Single.	Size.	Size.	Size.	Size.	Size.	Size.	Size.	Size.	Size.	Size
			Iron Strop Blocks.									
			Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
Throat halyards *a*	2	—	4½	5	6	7	8	9	9½	10	10½	11
Peak halyards ...	5	1	4½	5	6	7	8	9	9½	10	10½	11
Peak downhaul...	—	1	3½	4	4	4½	4½	5	5	5	5	5
Main sheet	2	—	4½	5	6	7	8	9	9½	10	10½	11
Main sheet lead	—	2	4	4½	5	5½	6	6½	7	7½	8	8½
Pendant	—	2	4	4½	5	5½	6	6½	7	7½	8	8½
Runner tackle *b* ..	2	2	4	5 & 4½	6 & 5	7 & 5½	8 & 6	9 & 6½	9½ & 7	10 & 7½	10½ & 8	11 & 8½
Outhaul............	—	1	3½	4	4½	5	5½	6	6½	7	7½	8
Topsail halyards.	—	1	3½	4	4½	5	5½	6	6½	7	7½	8
Topsail sheet ...	—	1	3½	4	4½	5	5½	6	6½	7	7½	8
Topmast stay purchase	1	1	3½	4	4½	5	5½	6	6½	7	7½	8
Topmast backstays	4	4	3½	4	4½	5	5½	6	6½	7	7½	8
Preventer backstays	2	2	3½	4	4½	5	5½	6	6½	7	7½	8
Preventer backstay whips......	—	2	3½	4	4½	5	5½	6	6½	7	7½	8
Throat purchase.	1	1	3½	4	4½	5	5½	6	6½	7	7½	8
Peak purchase ...	1	1	3½	4	4½	5	5½	6	6½	7	7½	8
Topping lifts......	—	4	3½	4	4½	5	5½	6	6½	7	7½	8
Topping lift purchase	2	2	3½	4	4½	5	5½	6	6½	7	7½	8
Spinnaker topping lift *c*	2	—	3½	4	4½	5	5½	6	6½	7	7½	8
			Rope Strop Blocks.									
Reef tackle *d* ...	1	1	5½ & 3½	7 & 4	7½ & 4½	8 & 5	8½ & 5½	9 & 6	9½ & 6½	10 & 7	10½ & 7½	11 &
Tack tackles......	3	3	3½	4	4½	5	5½	6	6½	7	7½	8
Spinnaker halyards............	- -	1	3½	4	4½	5	5½	6	6½	7	7½	8
Spinnaker guy whip	—	1	3	3	3½	4	4½	5	5½	6	6½	7
Topsail sheet whip	—	1	3½	4	4½	5	5½	6	6½	7	7½	8
Topmast staysail halyards	—	1	3	3½	4	5	5½	6	6½	7	7½	8
Topmast staysail tack	—	1	3	3½	4	5	5½	5	5½	6	6½	7
Topmast staysail sheet	—	1	3	3½	4	5	5½	6	6½	7	7½	8
Trysail sheets ...	2	2	5	5½	6	6½	7	7½	8	8½	9	9
Boom guy.........	1	1	3½	4	4½	5	5½	6	6½	7	7½	8
Dead-eyes *e*	—	12	3½	4	4½	5	5½	6	6½	7	7½	8

(*a*) The upper throat halyard block will be treble.
(*b*) Fiddle blocks.
(*c*) Single blocks under 75 tons.
(*d*) Fiddle blocks.
(*e*) Only two shrouds a-side under 40 tons.

Table X.—SCHOONERS.

FOREMAST.

Iron Strop Blocks.

B. Racer. C. Cruiser.	Description.		Tons B. 20. C. 30.	Tons B. 30. C. 40.	Tons B. 40. C. 60.	Tons B. 50. C. 75.	Tons B. 75. C. 100.	Tons B. 100. C. 140.	Tons B. 125. C. 180.	Tons B. 150. C. 220.	Tons B. 175. C. 250.	Tons B. 200 C. 275.
Name of Block.	Number. Double.	Number. Single.	Size.	Size.	Size.	Size.	Size.	Size.	Size.	Size.	Size.	Size.
			Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
Throat halyards	2	—	4½	5	6	7	8	9	9½	10	10½	11
Peak halyards	5	—	4½	5	6	7	8	9	9½	10	10½	11
Downhaul	—	1	3½	4	4½	5	5	5	5	5	5	5
Vang	—	1	3½	4	4½	5	5½	6	6½	7	7	7
Fore sheet..............	—	2	3½	4	4½	5	5½	6	6½	7	7½	8
Fore sheet purchases	2	2	3½	4	4½	5	5½	6	6½	7	7½	8
Pendant.................	—	1	4	4½	5	5½	6	6½	7	7½	8	8½
Runner tackle *a*	2	2	6½ & 3½	7 & 4	7½ & 4½	8 & 5	8½ & 5½	9 & 6	9½ & 6½	10 & 7	10½ & 7½	11 & 8
Topsail halyards	—	1	3½	4	4½	5	5½	6	6½	7	7½	8
Topsail sheets (cheek)	—	2	4	4½	5	5½	6	6½	7	7½	7½	8
Sheet lead..............	—	2	3½	4	4½	5	5½	6	6½	7	7½	8
Topmast backstays ...	4	4	3½	4	4½	5	5½	6	6½	7	7½	8
Preventer backstays ..	2	2	3½	4	4½	5	5½	6	6½	7	7½	8
Throat purchase	1	1	—	—	—	—	5½	6	6½	7	7½	8
Peak purchase	1	1	3½	4	4½	5	5½	6	6½	7	7½	8
Spinnaker topping lift	2	—	3½	4	4½	5	5½	6	6½	7	7½	8
Fore staysail sheets	—	2	3½	4	4½	5	5½	6	6½	7	7½	8
,, ,, purchases	—	4	3½	4	4½	5	5½	6	6½	7	7½	8
Fore staysail halyards	—	2	3½	4	4½	5	5½	6	6½	7	7½	8
Downhaul	—	1	3½	4	4	5	5	5½	6	6½	7	7
Bobstay.................	1	1	4½	5	6	7	8	9	9½	10	10½	11
Bowsprit shrouds......	2	2	6½ & 3½	7 & 4	7½ & 4½	8 & 5	8½ & 5½	9 & 6	9½ & 6½	10 & 7	10½ & 7½	11 & 8
Jib halyards	—	3	3½	4	4½	5	5½	6	6½	7	7½	8
Jib tack..................	—	1	3½	4	4½	5	5½	6	6½	7	7½	8
Jib purchase runner	—	1	—	—	—	5	5½	6	6½	7	7½	8
Jib purchase...........	1	1	3½	4	4½	5	5½	6	6½	7	7½	8
Dead-eyes..............	12	12	4	4½	5	5½	6	6½	7	7½	8	8½
						Rope Strop Blocks.						
Spinnaker halyards...	—	1	3½	4	4½	5	5½	6	6½	7	7½	8
Spinnaker guy whip...	—	1	3	3	4	4½	5	5½	6	6½	7	7
Trysail sheets	2	2	5	5½	6	6½	7	7½	8	8½	9	9½
Jib topsail halyards...	—	1	3½	4	4½	5	5½	6	6½	7	7½	8
Fore staysail sheets ..	2	2	3½	4	4½	5	5½	6	6½	7	7½	8
Jib sheets...............	—	2	—	—	4½	5	5½	6	6½	7	7½	8
Burton	—	2	3	3½	4	4½	5	5½	5½	6	6½	7
Downhaul (forward)..	—	1	3	4	3½	4	5	5	5	5	5	5

(*a*) The double blocks are fiddles.

Table XI.—SCHOONERS.

SIZES OF CORDAGE.

R. Racers. C. Cruiser. / Name of Rope.	Tons. R. 20. C. 30. Cf.	Tons. R. 30. C. 40. Cf.	Tons. R. 40. C. 60. Cf.	Tons. R. 50. C. 75. Cf.	Tons. R. 75. C. 100. Cf.	Tons. R. 100. C. 140. Cf.	Tons. R. 125. C. 180. Cf.	Tons. R. 150. C. 220. Cf.	Tons. R. 175. C. 250. Cf.	Tons. R. 200. C. 275. Cf.
	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
Main and fore halyards	1¾	2	2½	2¾	3	3¼	3½	3¾	4	4¼
Fore staysail halyards..	1¼	1½	1½	1¾	1¾	2	2¼	2½	2¾	3
Jib halyards *a*............	1¾ (3/16)	2 (3/16)	2¼ (¼)	2½ (5/16)	2¾ (5/16)	3 (⅜)	3¼ (½)	3½ (⅝)	3¾	4
Topsail halyards.........	1¼	1½	1¾	1¾	2	2½	2¾	3	3¼	3½
Topsail trip halyards...	1	1	1¼	1½	1½	1¾	2	2¼	2½	2¾
Jib topsail halyards ...	1	1¼	1¼	1½	1¾	1¾	2	2¼	2½	2¾
Maintopmast staysail halyards	1	1¼	1¼	1½	1¾	1¾	2	2¼	2½	2¾
Spinnaker halyards and topping lift; also fore vang.....................	1	1¼	1½	1½	1¾	1¾	2	2¼	2½	2¾
Main sheet	1½	1¾	2	2½	3	3¼	3½	3¾	4	4¼
Fore sheet	1¾	2	2½	2¾	3¼	3¾	4¼	4¾	5	5¼
Fore staysail sheets ...	1½	1¾	2	2½	3	3¼	3½	3¾	4	4¼
Jib sheets	2¼	2½	2¾	3	3¼	3½	3¾	4	4¼	4½
Topsail sheets............	1½	1½	1¾	1¾	2	2¼	2½	2¾	3	3¼
Jib topsail sheets	1	1¼	1¼	1½	1¾	1¾	2	2¼	2½	2¾
Maintopmast staysail sheets	1	1¼	1¼	1½	1¾	1¾	2	2¼	2½	2¾
Spinnaker sheets.........	1	1¼	1¼	1½	1¾	1¾	2	2¼	2½	2¾
Trysail sheets	1¾	1¾	2	2¼	2½	2¾	3	3¼	3½	3¾
Jib tack	2	2¼	1¼ *w*	1½ *w*	1¾ *w*	2 *w*	2¼	2½ *w*	2¾ *w*	3 *w*
Tack tackles	1¼	1¼	1½	1½	1¾	1¾	2	2¼	2½	2¾
Main, fore, and jib purchases	1¼	1¼	1½	1½	1¾	1¾	2	2¼	2½	2¾
Downhauls	1	1¼	1¼	1½	1¾	1¾	2	2	2	2
Burton.....................	1	1¼	1½	1¾	2	2¼	2½	2¾	3	3¼
Bowsprit shrouds tackle	1¼	1¼	1½	1½	1¾	1¾	2	2¼	2½	2¾
Fore sheet purchases ...	1¼	1¼	1½	1½	1¾	1¾	2	2¼	2½	2¾
Bobstay tackle	1½	1¾	2	2½	3	3½	3¾	4	4¼	4½
Runners (wire)	1¼	1½	1¾	1¾	2	2¼	2½	2¾	3	3¼
Runners (hemp or manilla)	2	2¼	2½	2¾	3	3¼	3½	3¾	4	4¼
Runner tackles	1¼	1¼	1½	1½	1¾	1¾	2	2¼	2½	2¾
Topping lifts	1¼	1¾	2	2½	3	3½	3¼	4	4¼	4½
Topping lift purchase...	1¼	1¼	1½	1½	1¾	1¾	2	2¼	2½	2¾
Reef pendants............	1½	1¾	2	2½	3	3½	3¾	4	4¼	4½
Reef tackle	1¼	1¼	1½	1½	1¾	1¾	2	2¼	2½	2¾
Boom guy	1½	1¾	2	2½	3	3½	3¾	4	4¼	4½
Boom guy purchase ...	1¼	1¼	1½	1½	1¾	1¾	2	2¼	2½	2¾
Lanyards..................	1¾	2	2¼	2½	2¾	3	3¼	3½	3¾	4

The remarks on page 43 apply equally to schooners, and the tables of blocks and cordage in Tables IX., X., XI., will be suitable for schooners rigged pretty much as those mentioned in Table XII. are.

(*a*) The small figures in brackets are the size of the links for chain halyards.

(*w*) Wire rope.

TABLE XII.—DIMENSIONS OF YACHTS AND SPARS.

Name, with dates when the spars were carried.	Rig.	Length of L.W.L.	Extreme Beam.	Extreme draught of water.	Length of Mainmast, Deck to Hounds.	Length of Bowsprit outside Stem Head.	Length of Main Boom, Mast to Pin of Outhaul.	Length of Main Gaff.	Length of Topmast Fid to Halyard Sheave.	Area of Mainsail.	Area of Head Sail, Y.R.A.	*Total Sail Area, Y.R.A.
		FT.	FT.	FT.	FT.	FT.	FT.	FT.	FT.	SQ. FT	SQ. FT	SQ. FT
Amphitrite (1890) ..	S	94·5	19·4	...		...	64·0	37·5	...	3158	2618	8297
Waterwitch (1887)...	S	95·7	19·3	12·2	58·0	33·0	62·0	36·0	37·0	2868	2649	8090
Egeria (1880)	S	93·7	19·2	12·5	58·0	32·0	66·0	38·5	35·0	3040	2600	8500
Pantomime (1876) ...	S	91·5	19·3	12·0	57·5	31·5	58·0	35·0	34·0	2622	2350	7870
Miranda (1879)	S	86·7	18·9	13·0	58·8	27·5	64·8	35·5	34·0	3075	2136	7700
Dracæna (1876) *b* ...	S	80·0	18·0	10·0	52·0	28·0	48·0	30·0	30·0	2040	2008	5920
Flying Cloud (1872)..	S	73·5	15·7	9·5	46·0	23·5	46·5	28·0	28·0	1690	1920	5500
Latona (1882).........	Y	93·6	20·2	12·5	58·3	39·0	63·0	48·0	48·5	3390	3900	8880
Florinda (1878)	Y	85·7	19·3	11·9	54·5	36·0	56·5	42·5	44·0	2923	3300	8280
Jullanar (1877)	Y	99·0	16·9	13·8	53·0	24·5	56·5	42·0	38·5	2737	2900	7800
Constance (1885) *b*...	Y	82·8	18·2	12·0	49·0	28·5	53·0	37·5	39·0	2330	2600	6190
Lethe (1890)	Y	93·3	19·6	...	...	...	63·2	40·3	...	2936	3454	7958
Caroline (1878) *b* ...	Y	75·0	16·1	11·5	43·5	31·0	50·7	37·0	35·0	1950	2340	5050
Foxglove (1890)......	Y	61·1	13·9	...	37·8	23·0	43·2	30·5	34·5	1451	1592	3919
Oimara (1872)	C	95·0	19·9	13·0	64·0	46·0	72·0	49·0	49·5	3960	4500	9520
Thistle (1887)	C	86·4	22·2	...	...	35·0	81·4	51·5	...	4563	3770	9957
Britannia (1896)	C	87·8	23·6	...	64·0	...	91·0	55·0	...	5165	3889	10057
Meteor (1896)	C	89·0	24·3	...	69·0	...	96·6	58·7	...	5910	4718	12327
Isolde (1896)	C	60·0	17·0	11·5	41·5	...	60·5	34·0	...	2110	1292	4006
Saint (1896)	C	46·9	12·2	...	33·2	...	51·5	28·1	...	1508	1026	2976
Satanita	C	98·1	24·7	16·3	62·0	...	89·3	54·0	...	4972	3802	10048
Vigilant (1895)	C	87·3	26·0	14·0	...	...	96·7	53·5	...	5695	4397	11588
Iverna (1890)	C	83·5	19·0	13·0	...	...	77·9	47·3	...	3856	3410	8458
Valkyrie (1890)	C	69·2	15·9	...	...	...	69·0	42·2	...	3047	2736	6707
Yarana (1890)	C	65·7	14·9	...	...	...	62·7	40·6	...	2624	2276	5651
Vanduara (1890) ...	C	81·3	16·2	12·4	48·5	31·0	68·3	43·3	42·7	3170	2930	7283
Genesta (1887)	C	81·0	15·0	13·0	52·0	35·0	70·0	46·0	47·5	3090	3360	7646
Marjorie (1884)	C	75·4	14·5	...	...	...	...	...	...	2964	2928	7022
Annasona (1883)......	C	64·3	11·9	10·7	41·0	31·0	55·0	37·0	38·0	2130	2104	4986
Creole (1890)	C	59·3	13·3	12·1	...	...	56·1	34·0	...	1881	1570	4008
Carina (1895)	C	60·8	15·8	12·8	...	...	60·0	35·3	...	2167	1306	3947
Vendetta (1894)	C	60·5	17·1	11·8	...	...	61·0	34·5	...	2151	1282	3963
Tara (1883)............	C	66·0	11·5	11·5	42·5	30·0	58·0	39·5	40·0	2270	2150	5280
Freda (20 tons) (1881)	C	49·0	9·8	9·5	34·5	24·0	43·0	28·0	30·0	1450	1480	3150
Vanessa (20 tons) ('78)	C	47·0	9·8	7·8	31·5	24·0	39·0	27·5	26·5	1150	1170	2720
Vreda (1889)	C	45·4	10·1	...	34·0	*c* 21·0	42·6	26·0	...	1230	1024	2641
Stephanie (1895) ...	C	46·7	12·3	10·9	33·0	10·0	49·0	28·8	23·4	1497	689	2565
Dragon III. (1894)...	C	46·1	13·2	9·0	...	...	47·0	27·0	...	1354	874	2600
Audrey (1895)	C	44·03	13·16	9·0	...	...	48·5	27·0	...	1403	976	2740
Asphodel (1894)......	C	46·6	12·3	9·1	...	...	46·2	27·2	...	1352	876	2576
Luna (1894)	C	46·1	13·0	9·7	...	...	47·8	26·5	...	1348	916	2599
Ulerin (1884)	C	41·5	7·2	...	29·0	19·0	38·5	25·0	28·0	1040	1049	2492
Decima (1889).........	C	35·7	10·2	8·5	31·0	...	34·5	23·8	...	905	497	1679
Lilith (1894)	C	35·2	10·6	7·5	...	...	38·0	21·5	...	866	568	1695
Archee (1890).........	C	30·4	9·6	...	...	...	30·6	19·5	...	675	305	980
Flat Fish (1894) ...	L	32·0	10·0	7·5	...	...	31·8	32·4	...	687	232	919
Gareth (1894).........	L	28·9	6·9	6·7	...	...	24·9	24·5	...	442	97	538
Meneen (1894)	L	24·8	7·0	6·0	...	...	27·2	27·0	...	474	123	603
Humming Bird (1890)	L	25·9	7·3	5·7	25·5	7·0	23·5	24·2	...	432	135	567
Babe (1890)	L	26·8	6·8	5·9	...	...	23·6	22·8	...	418	135	553
Dolphin (1890)	L	25·7	7·4	5·8	...	...	23·5	23·5	...	428	152	580
Doris (1890)	C	33·62	5·73	6·3	21·5	...	35·0	21·3	...	782	686	1730
Currytush (1885).....	C	28·53	4·7	5·5	23·0	...	26·2	18·0	..	539	422	1047

* This includes topsails. Several accurate sail plans will be found among the plates.
b Cruiser. *c* From fore end of L.W.L. to cranse iron.

SPECIFICATIONS FOR BLOCKS OF A CUTTER YACHT OF ABOUT 85FT. RATING.

By Mr. John Harvey.

FOR SPARS AND RIGGING.

Purpose.	Internal Iron Bound. No.	Internal Iron Bound. Description.	Internal Iron Bound. Inches.	Internal Iron Bound. Size of Rope.	Rope Strop. No.	Rope Strop. Description.	Rope Strop. Inches.	Rope Strop. Size of Rope.	Total No.	Details.
Forestay harts	1	Oval	9	2¼	1	Oval	9	2¼	2	One iron bound with lugs and ⅞in. pin. One scored for 3¾in. wire rope.
Rigging deadeyes	8	Round	5¾	3¼	8	Round	5¾	3¼	16	Eight with iron bindings to fit chain plates. Eight scored for 3in. wire rope.
Cat block	1	Double	7	2½	—	—	—	—	1	With large hook.
Snatch block	1	Single	12	4½	—	—	—	—	1	Iron bound, with jointed clasp and large hook to swivel.
Main tackles pendant	2	Single	7½	3½	—	—	—	—	2	Eye and thimble to take 3in. wire rope. Spoon hook for runner.
Main tackles on runner	2	Fiddle	12½	2¾	—	—	—	—	2	Eye and thimble to take 3⅝in. rope.
Main tackles	2	Single	6½	2¾	—	—	—	—	2	Spoon hook, to eye at rail, becket.
Bobstay	1	Double	—	3¾	—	—	—	—	1	With shank 20in. long, eye at end 3⅛in. by 1⅝in. the clear.
Bobstay	1	Single	—	3¾	—	—	—	—	1	Double lugs to take ⅝in. chain, becket to be clenched in hawse of block.
Bobstay preventer	— / —	— / —	— / —	— / —	1 / 1	Double / Single	— / —	3 / 3	2	Double to have tail. Single to take shackle.
Bobstay up-haul	1	Single	3	2	—	—	—	—	1	Solid eye for lashing. Sheave 2½in. over.
Bowsprit heel rope	1	Single	8	3½	—	—	—	—	1	Mouse hook.
Bowsprit shrouds	2	Single	7	3	—	—	—	—	2	Solid eye for shackle. Shackle to reeve through shoulder eye in deck.
Bowsprit shrouds	2	Fiddle	10½	3	—	—	—	—	2	Solid eye to take shoe with 1⅛in. pin.
Topmast backstays	6	Double	6	2½	—	—	—	—	6	Solid eye for shackle. Shackle to reeve through thimbles when backstay (1⅜in.) is in place, and served .... } All to have beckets.
Topmast backstays	6	Single	6	2½	—	—	—	—	6	Four with solid eyes for shackles to eyes in channels, two to have spoon hooks .... } All to have beckets.
Topmast extra tackle	—	—	—	—	2	Single	5	2	2	With tail.
Topmast heel rope	— / —	— / —	— / —	— / —	1 / 1	Double / Single	6	3	2	Mouse hook on the double and tail on the single.
Topmast stay tackle	—	Single	4	2	—	—	—	—	1	Shank 13in. long, and double lugs ¾in. in the clear, and ⅞in. pin.

FOR SAILS.

Purpose.	Internal Iron Bound.				Rope Strop.				Total No.	Details.
	No.	Description.	Inches.	Size of Rope.	No.	Description.	Inches.	Size of Rope.		
Main halyard	1	Treble	9½	3½	—	—	—	—	1	To main halyard bolt, shackle (to pass through bolt), patent sheaves.
Main halyard	1	Double	9½	3½	—	—	—	—	1	To toggle in gaff, shackle (to pass through bolt), patent sheaves.
Main halyard purchase	2	Double	7½	2½	—	—	—	—	2	Lower one to have mouse hook (no swivel), the upper one with eye to take main part of halyard and a becket, both patent sheaves.
Gaff halyards	5	Single	9½	3½	—	—	—	—	5	All sister hooks, patent sheaves.
Gaff halyards purchase	2	Double	7	2½	—	—	—	—	2	Lower one bound the same as main purchase, upper one with eye to take 3½in. rope, both to have patent sheaves.
Main sheet	—	—	—	—	1	Treble	10	3½	1	Double wire strop to boom.
Main sheet	1	Double	10	3¾	—	—	—	—	1	Solid eye binding, with shackle to take (and which is to run on) hawse, the ropes to pass through the block vertically.
Main sheet lead	2	Single	7½	3¾	—	—	—	—	2	Clump, eye to take (and pass through) eye bolt, and stock of eye bolt to be left 15in. long.
Fore halyards	2	Single	6½	2¼	—	—	—	—	2	Sister hooks, the hooks of lower one to take head of sail.
Fore tack	—	—	—	—	1	Double	5½	2¼	1	The double scored for wire tie, and the single to be rope strop leathered, with mouse hook. A spoon hook to take thimble of sail.
	—	—	—	—	1	Single	5½	2	1	
Fore sheets	—	—	—	—	2	Single	7	3¾	2	Wire strop with cringle in centre for grommet.
Fore sheets tackles	2	Single	5½	2	—	—	—	—	2	Hook to mouse, a becket in each one, to hook to eye in stanchion.
Fore downhaul	—	—	—	—	2	Double	5½	2	2	With score to take fore sheet runner, 3¾in.
	—	—	—	—	1	Single	5	2	1	Leathered, and tail to seize on to stay.
Jib halyards	3	—	—	—	—	—	—	—	3	All clip hooks, the lower block large ones to take head of sail; sheaves selected to take large ¾in. chain.
Jib halyards purchase	1	Treble	7	2½	—	—	—	—	1	The treble to shackle on to chain, the double to hook to eye in deck (no swivel) and to have a becket, both to have patent sheaves.
	1	Double	7	2½	—	—	—	—	1	
Jib outhaul	1	Single	7½	3½	—	—	—	—	1	Eye and thimble to take wire tye.
Jib downhaul	—	—	—	—	1	Single	6	2	1	With tail end.
Mainsail outhaul	1	Fiddle	10	2½	—	—	—	—	1	Solid shackle to take chain outhaul, the inner end to be a clump on the boom.

FOR SAILS—*continued.*

Purpose.	Internal Iron Bound. No.	Internal Iron Bound. Description.	Internal Iron Bound. Inches.	Internal Iron Bound. Size of Rope.	Rope Strop. No.	Rope Strop. Description.	Rope Strop. Inches.	Rope Strop. Size of Rope.	Total No.	Details.
Mainsail truss	1	Double	5	2	—	—	—	—	1	The solid eye under the horn of gaff to be chased for middle plate in block.
Mainsail truss	—	—	—	—	1	Single	5	2	1	With tail to seize to mainsail (thimble).
Mainsail tack	—	—	—	—	1	Double	6	2¼	1	With tail to go through thimble in mainsail.
	—	—	—	—	1	Single	6	2¼	1	Leathered strop, mouse hook.
Mainsail reef tackle	1	Fiddle	12	2½	—	—	—	—	1	Large hook (to mouse) to take reef earings.
	1	Single	6½	2½	—	—	—	—	1	With becket and mouse hook to span hoop on boom.
Main boom guy	—	—	—	—	1	Double	7	2¾	1	Leathered strop; the double to have large hook to take leathered strop on boom or pendant, the single to hook to eye plate on channel.
	—	—	—	—	1	Single	7	2¾	1	
Topping-lifts	2	Single	8½	8¾	—	—	—	—	2	Clip hooks (short and stout) to eye plates on cheeks.
Topping-lift runner	2	Single	7	3	—	—	—	—	2	Eye to receive topping-lift, 3¾in. rope.
Topping-lift purchase	2	Single	6	2¼	2	Single	6	2¼	4	Hook (mouse) to eye in channels.
Peak downhaul	1	Single	5	2	—	—	—	—	1	Bound into eye, to drive into gaff.
Topsail halyards	1	Single	6	3	—	—	—	—	1	Short stout sister hook, no purchase.
Topsail tack	—	—	—	—	1	Double	6	2½	1	Leathered strops.
	—	—	—	—	1	Single	6	2½	1	
Topsail sheet	1	Single	6	3½	1	Single	7	3½	2	Plates to go to eye under gaff horns, and the other to have a pendant.
Topsail clueline	—	—	—	—	1	Single	5	2	1	With tail rope.
Trysail sheets	—	—	—	—	4	Double	8	3	4	Strops leathered.
Burton	—	—	—	—	2	Single	6	2¼	2	For rope strops.
Handy Billy	—	—	—	—	2	Double	6	2½	2	Rope strops leathered.
	—	—	—	—	2	Single	6	2½	2	
Tiller rope	—	—	—	—	2	Single	5½	2¼	2	Swivel sister hooks, to eye in rail.
Spinnaker halyards	—	—	—	—	2	Single	6	2½	2	Tail to one, and eye over pole to topmast to the other
Spinnaker guy (fore)	—	—	—	—	2	Single	5	2	2	Tails.
Spinnaker guy (main)	—	—	—	—	2	Single	6	2½	2	Tails.
Spinnaker Burton	—	—	—	—	1	Double	6	2¼	2	Tails.
	—	—	—	—	1	Single				
Jib topsail halyard	—	—	—	—	1	Single	5	2¼	1	Tail.
Jib topsail tack	—	—	—	—	2	Single	5	2	2	Tails.
Spare blocks or extra tackle	—	—	—	—	8	Single	5½	2¼	8	Mouse hooks.
	—	—	—	—	8	Double	5½	2¼	8	Tails.
Spare blocks for topmast stay, &c.	—	—	—	—	8	Double	5	2	8	Tails.
	—	—	—	—	8	Single	5	2	8	Mouse hooks.
Snatch blocks	—	—	—	—	2	Single	9	8	2	Tails.

CHAPTER XIX.

BALLAST.

The displacement of a vessel is a quantity which enters very largely into any consideration of her stability, as was abundantly shown in a former chapter, and in no way does the knowledge of a yacht's displacement more largely assist the naval architect in his labours than in the matter of ballast. Most English yachts are built of pretty much the same scantling, and have similar internal fittings and spars; yet the weight, exclusive of ballast, of any two yachts of equal length and breadth may vary considerably, as will be gleaned from the following table:

Name of Yacht.	Weight of Ballast in Tons.	Weight of this Ballast on Keel.	Ratio of Ballast to Displacement.	Displacement in Tons.
Egeria (wood)	60	30	·422	142
Florinda (wood)	52	23*	·361	144
Miranda (wood)	78	30	·488	160
Constance (wood)	61	28·5	·452	141
Erycina (wood)	71	57	·534	133
Galatea (iron)	78	78	·493	158
Ailsa (composite)	63·1	62·1	·404	156
Genesta (composite)	72	72	·510	141
Milly (composite)	42	6	·508	82·7
Isolde (composite)	22·2	22·2	·483	46
May (composite)	30	28	·500	60·5
Sleuthhound (wood)	37	34	·585	63·2
Tara (composite)	41	37	·545	75·3
Castanet (composite)	30	28	·500	60
Clara (wood)	21·9	all	·600	36·5
Vanessa (wood)	16·5	2·5	·579	28·5
Ghost (wood)	15·9	all	·521	30·5
Penitent (wood)	17	17	·634	26·8
Vreda (steel)	14·45	all	·500	28·9
Dragon (wood)	15·0	all	·574	26·1
Minerva (wood)	12·4	all	·590	21 0
Neptune (wood)	8·5	8·5	·566	15
Saraband (wood)	12	11·5	·571	21
Drina (wood)	7·0	6·6	·466	15·0
Dis (wood)	8·75	all	·591	14·8
Olga (composite)	7·2	6·9	·681	10·5
Oread (wood)	4·8	all	·657	7·3
Lady Nan (wood)	2·5	all	·610	4·1
Dolphin (wood)	2·1	all	·555	3·6

* Florinda originally only had 3½ tons of lead on her keel; but this was subsequently increased to 8 tons, and before her last racing season a lead keel was cast broader than her wood keel, and lead slabs bolted on above through the wood keel at *a a*.

It is very evident that, the greater quantity of ballast a vessel has in proportion to her displacement, the lower ought to be her centre of gravity, and, as a sequence, the greater ought to be her stability. A yacht builder, therefore, even if he disregards the simple calculation of the displacement of a vessel he is commissioned to construct, will know that, so far as sail-carrying power is concerned, an advantage must accrue from making the fabric of the hull as light as possible, consistent with security, if the vessel be intended for racing.

But it must not be concluded that of two vessels of equal length, breadth, and draught of water, and equal scantling so far as sizes of timbers, beams, and planking go, the weight of the hull need be the same. The probability is that the vessel of the larger displacement will carry the greater proportionate weight of ballast, as is very pointedly exemplified in the case of the Egeria and Miranda; the scantling of the two is about equal so far as thickness of plank and siding of frame go; yet, in proportion to displacement, it will be seen that the Egeria carries a smaller quantity of ballast. The reason of this is dependent upon the different forms taken by the vertical sections in the two vessels. The Miranda's midship section is of the peg-top form, and the greatest girth of the midship frame, from the load line to the keel, is 14ft.; the Egeria has a hollower section, very flat from the bilge to the garboard, and then very curved. The consequence is that the girth or length of the frame, from the load line to the keel, is 16ft. and there will be a proportionate excess of length in nearly every frame, and, as a consequence, an excess of plank as well. For this reason, unless vessels of similar linear dimensions be also of similar form, they need not have the same ratio of ballast to displacement. It will be noticed in the table that as the tonnage size of the yachts decreases the ratio of ballast to displacement increases; the cause of this is that depth in a small yacht is relatively greater than in a large one—or, in other words, the depth is not made to decrease in proportion to the decrease in the other two dimensions of length and breadth —and as the weight of material used in the construction is very much less in proportion to the displacement, as a consequence more ballast can be carried. So far as lightness of hull goes for any given form there is some advantage on the side of the composite construction, such as May, Milly, Tara, &c., have.

A very great change has been wrought in the manner of ballasting yachts since 1880, and a better type of vessel for speed and weather lines has been built since ballasting has received greater attention. Scrap iron, fire bars, and even stone and shingle, were the common form of ballast in the first days of yachting, with a few shot bags to trim to windward in the event

of match sailing. As the pastime of yachting developed, a taste for yacht racing was very rapidly disseminated, and ballast shifting came to be regarded as part of the science of the sport; and thus success, in fair sailing breezes, frequently rewarded the man who had the greatest weight of shot bags, and the greatest number of men to shift them. There is no denying that the absence of restrictions upon the shifting of ballast was introducing us to a very bad type of vessel, and the thanks of the present generation are due to the men who in 1856 placed a stringent prohibition upon the practice of trimming ballast to windward. It was made a point of honour with yacht owners that they would not permit ballast to be shifted during a race.

When the prohibition of shifting ballast came into operation, it was only reasonable that yachtsmen and yacht builders should devise some plan to enable the yachts to continue to carry the enormous spars and sail areas that shifting ballast had called into existence. The shot bags that had been shifted from bilge to bilge were permanently located in a solid form in the bottom of the hull, and metal keels—generally of iron—became fashionable. Then lead keels were timorously introduced. Indeed, lead ballast generally was contemned, and when a yacht first appeared with "all lead ballast," many so little understood what its effect would be as to predict that the yacht with a lead keel would "get caught some day, and sink like a stone." This ludicrous prediction, so far as we know, has not been verified, and "all lead" for racing vessels is now commonly employed as ballast, and 99 per cent. of it on the keel outside, and frequently the whole weight. There is thus no limit to the proportion of ballast to be carried outside, and, whilst the general practice up to 1878 had been to put about one-fourth of the ballast outside, on the keel, or in the garboards, the practice now is as stated above in racing yachts; whilst a cruiser might have two-thirds of the weight outside. In steel-built vessels, such as Vanduara and Galatea, the lower part of the hull or keel is made a kind of ballast box, and so no lead is actually carried "outside," the whole being run into the keel.

The principal advantage of introducing lead as ballast in yachts was that, owing to its greater specific gravity or smaller bulk for any given weight, it can be stowed as a whole in a lower position in the hull than a similar weight of iron, and thus bring about a lower position of the centre of gravity of the vessel, which simply means that, with any given weight, greater stability can be acquired by using lead as ballast than by using iron. Next, as the centre of gravity of a vessel can be brought so much under command by the use of lead and lead keels, advantage has been taken of this to decrease the displacement. In this respect the employment of lead as ballast has had a beneficial effect. It has enabled

naval architects to design vessels equal in accommodation to those of much greater displacement and at the same time to be more than their equal in initial stability.

In order to get the centre of gravity of the ballast as low as possible, the practice was, in most racing yachts, though not in cruisers, to spread the weights out in a fore-and-aft direction. For smooth-water sailing nothing but clear gain, so far as stability is concerned, will result from having the weights well distributed in a fore-and-aft direction; but the case will be different among waves, no matter what the ballast be (see page 58). The longer the radius of gyration, and the greater the metacentric height are made, the more violent will be the pitching and scending motions among waves, and the vessel will dive, and be very wet and uneasy. But, on the other hand, if the weights are much concentrated, like the short lead keels of modern racing yachts, the vessel, whilst

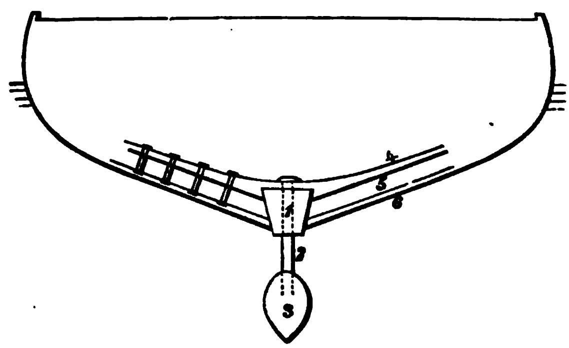


Fig. 230.

1. American elm keel.
2. Steel rod passing through keel.
3. Fish keel.
4. Iron timbers.
5. Frame of vessel.
6. Skin.

diminishing the momentum that caused her to plunge into a wave or "dive," will be "lively." The vessel with the longer radius of gyration will rise to the waves, and then bury herself. In moderate weather, and among comparatively small waves, the case might be somewhat different, as the momentum acquired would be unimportant, and the fore-and-aft motions of a vessel with a long radius of gyration would consequently be slow; whereas, a vessel with her weights much concentrated would tend to keep time with the waves, and the sails would not "sit" quietly in light winds. This is particularly noticeable in little boats, and often among *small* waves better results are obtained by stowing the weights well fore-and-aft. As a general rule, the best place for the ballast is in the middle third of the length of the vessel; and if two-thirds, more or less, of the ballast takes the form of a lead keel, the latter should not

be greater in length than four-sixths of the length of the vessel on the load water-line.

Various ingenious methods of ballasting have from time to time been adopted or proposed, and we have seen it suggested that ballast should be slung, or that it should have elastic bearings. It need scarcely be pointed out that moving-ballast among waves would be a very bad thing, as the momentum acquired by the ballast would in effect augment the general momentum of the vessel; and further, if the ballast were "slung," it would shift to leeward upon the vessel being heeled by a wind force, and it would be in effect like trimming ballast to leeward instead of to windward. A plan was also proposed in 1870 to suspend the lead keel by steel bars, as shown in Figs. 230 and 231. The obvious objection to this plan is that the bars would have to be of the greatest width transversely to withstand the lateral strain, and would thus offer a surface of much greater resistance than would arise from the whole of the spaces between the two keels being filled in solid, like the modern fin bulb keel.

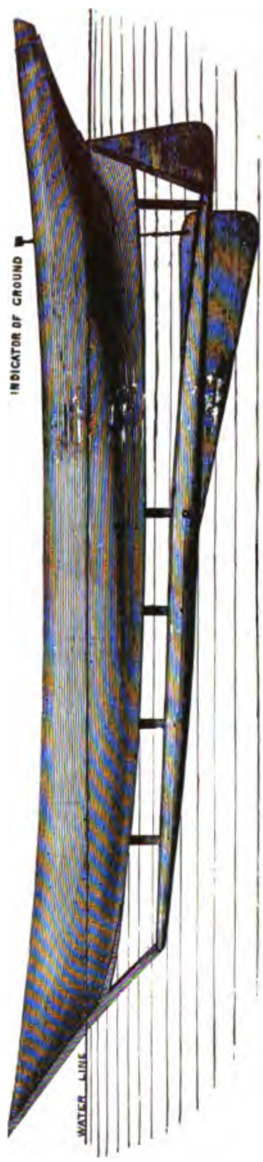


FIG. 231.

An elaborate plan in the way of ballasting is that introduced by Messrs. Harvey and Pryer; it consists of lead floors with angle iron inserted; lead keelson, with lumps of lead underneath it between the floors moulded to fill the spaces, and a lead outside keel. This plan was successfully tried in the Seabelle and other yachts, but the fashion of placing nearly the whole of the lead outside has rendered the plan obsolete.

About the year 1870 a plan was adopted of running the lead ballast into the vessel molten; and certainly this plan so far fulfilled its object, that the ballast was solid, and all the small spaces were filled. But the plan was objected to for the reason that if anything went wrong with the frame of the vessel, or if the ballast required shifting, it was such a very great labour to cut the solid mass out, and if the lead got in the seams it was apt to cause leakage. However, the practice is still followed with advantage in steel

"EVOLUTION"

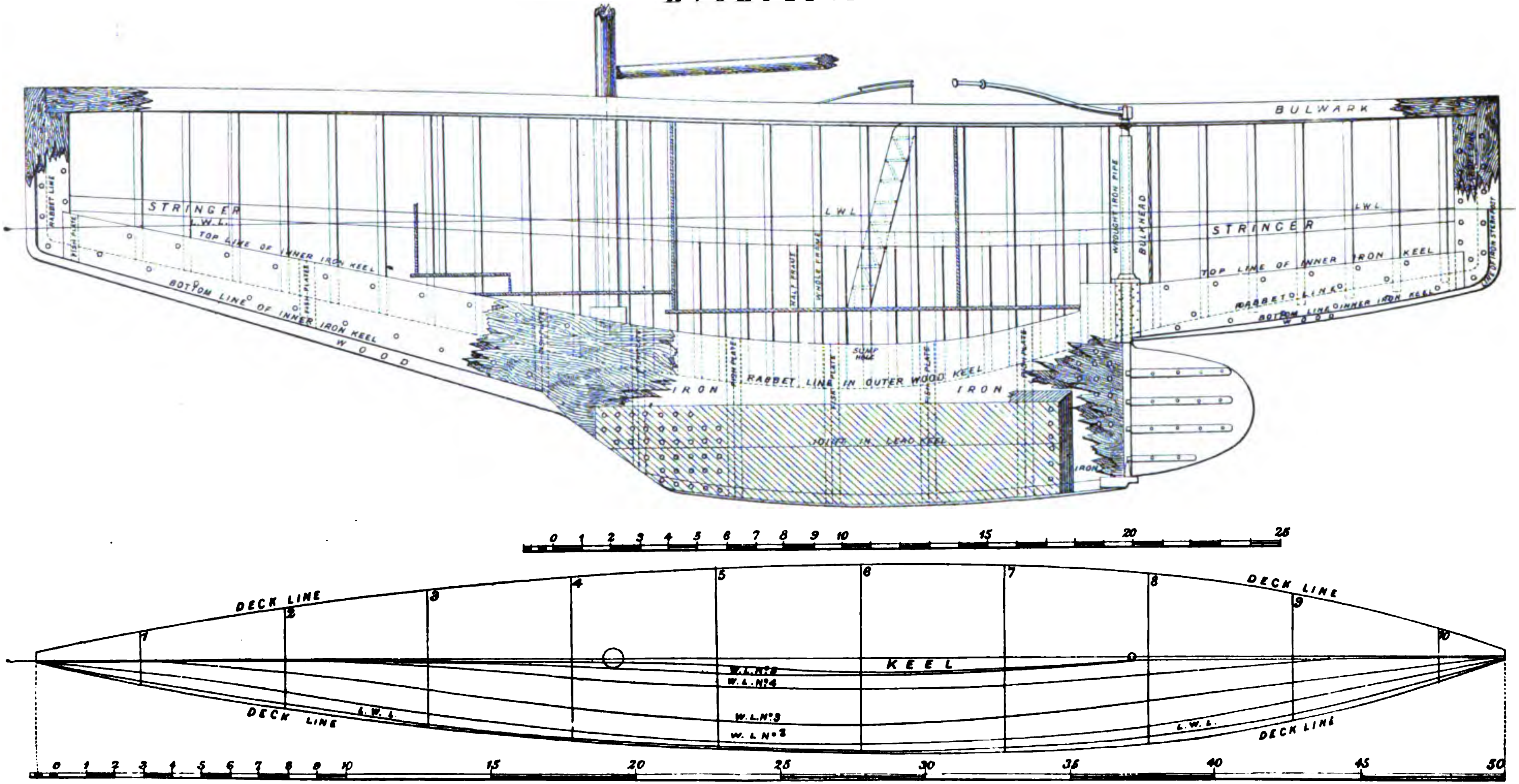


Fig. 282.

yachts, the keels of which form "ballast boxes." The Vanduara, Vreda, and Galatea were so ballasted, as already explained.

Sometimes the spaces between the heels of the timbers and the keel are filled with concrete (made of shot or boiler punchings and cement) up to the level of the top of the floors or keelson. If this plan is adopted, all the surfaces of the spaces should be well tarred, and the concrete should be thoroughly worked into every crevice, otherwise the damp may rot the heels of the frames and the plank, or foul bilge water may accumulate.

The modern plan of shaping lead keels is shown on the plates representing Isolde, Audrey, Flat Fish, Edie, &c., besides numerous woodcutsin the text (see page 447, &c.).

The object of making the underside of the keel semicircular is to obtain a minimum of surface for any given weight. Also, if weeds grow on the under side of the keel, they could easily be removed by working a rope fore and aft as the vessel lay afloat. Of course, if the lead keel were made rectangular the weight would be lower, but there would be an increase of surface. The objections to a very broad keel, and the advantages of a thin keel, are discussed on page 145.

These very thin lead keels now usually take the form of a "fin," as shown by the plate representing the Dolphin. This form of lead keel was introduced in 1880 by Mr. E. H. Bentall on a 10 tons yacht he named Evolution (see Fig. 232). The yacht was not a success; but that was mainly because she was of insufficient beam and displacement. Mr. Bentall then reduced the thickness of the fin, and placed an equivalent weight on it, as a bulb; we believe this was the first fin bulb keel fitted to a yacht. The form of lead keel was quite a sound conception, and was much recommended for small yachts as a "fixed centre plate" (in 1880). In 1887 Lieutenant Gartside Tippinge, R.N., fixed such a keel to his centre board yacht Mischief, and since then the idea has been successfully carried out in the small classes for racing on the Solent, Clyde, &c. Evolution was subsequently fitted with a bulb, and no doubt she was the first "fin bulb" boat built.

APPENDIX.

ELEMENTS OF WELL-KNOWN YACHTS, SOME OF WHICH ARE ILLUSTRATED BY THE TABLES OR REFERRED TO IN THE TEXT.

(The area of immersed surface and area of lateral resistance do not include the area of centre-boards.)

Yacht.	Rig.	Length on L.W.L.	Beam extreme.	Draught Water extreme.	Displacement.	Area mid section.	C.B. aft centre length.	Area immersed surface.	Area lateral resistance.	Sail Area, Y.R.A. rule.
		FT.	FT.	FT.	TONS.	SQ. FT.	FT.	SQ. FT.	SQ. FT.	SQ. FT.
Genesta	C	81·0	15·0	13·3	141·0	109·0	1·9	1940	864	8030
Samœna	C	80·7	16·2	13·0	130·0	102·0	3·5	2022	...	7350
Vanduara	C	81·2	16·2	12·3	130·0	98·0	2·0	1892	...	7283
Formosa	C	82·0	17·0	12·5	130·0	100·0	2·0	...	852	7120
Kriemhilda	C	79·5	17·5	12·1	115·4	89·7	1·3	2112	848	6792
Arrow	C	79·2	18·7	11·5	108.0	89·0	3·2	1940	760	6834
Lulworth	C	68·7	16·5	10·5	63.0	62·0	3·4	1414	539	5580
Castanet	C	58·8	14·0	11·5	60·0	66.0	...	1095	469	4067
May	C	64·0	12·0	11·0	61·0	...	...	1260	...	4990
Vanessa	C	47·0	9·8	7·8	28·5	36·	1·7	770	334	2720
Ghost	C	46·1	10·7	9·8	30·5	45·0	1·4	699	299	2577
Vreda	C	45·4	10·1	...	28·9	...	1·8	695	...	2641
Dragon	C	45·4	10·2	...	26·1	...	1·6	670	..	2632
Chiquita	C	45·5	11·1	9·6	28·0	...	...	...	303	2636
Minerva	C	40·5	10·2	9·0	21·0	33·0	1·1	572	239	3200
Saraband	C	41·0	7·3	7·5	21·0	31·6	1·7	578	257	2535
Drina	C	33·2	9·8	8·4	15·0	27·0	0·6	465	202	1801
Neptune	C	39·5	7·6	7·0	15·0	22·6	2·1	540	227	2150
Bedouin (cntre. brd.)	C	28·0	9·3	3·5 & 7	6·7	7·6	0·6	225	85	1016
Oread	C	28·2	7·0	6·6	7·3	17·0	1·1	302	128	1269
Mascotte	C	28·3	4·7	5·7	7·0	17·7	1·2	270	126	1050
Quinque	C	31·6	8·9	6·6	8·6	16·2	0·7	330	156	950
Valentine	C	29·5	7·0	7·4	7·0	16·0	0·9	319	135	997
Lady Nan	C	23·0	8·3	5·7	4·1	12·0	0·7	218	82	654
Dolphin	C	25·7	7·4	5·8	3·6	11·0	0·8	195	70	417
Lenore	Y	89·5	17·3	12·6	145.0	101·5	3·5	2223	862	7845
Constance	Y	82·8	18·1	12·0	141·0	107·0	2·2	1990	812	6190
Florinda	Y	85·9	19·3	11·8	150·0	106·0	1·3	2200	901	8280
Latona	Y	93·6	20·3	12·7	167·0	104·0	4·1	2480	980	8880
Jullanar	Y	100·0	16·8	13·7	158·0	106·0	0·4	2178	934	7800
Milly	Y	66·5	14·7	11·3	82·7	79·0	1·2	1515	638	4620
Sappho	S	122·0	27·0	13·0	232.0	124·0	6·5	3790	1355	14,243
Lyra	S	125·5	25·2	13·3	332.0	159·0	2·8	3450	1484	10,800
Seabelle	S	90·5	19·0	12·0	155.0	102·0	1·7	2330	1027	6880
Miranda	S	86·4	18·9	13·0	160·0	109·0	2·3	2228	925	7700

LENGTH OF THE CIRCULAR ARC SUBTENDED BY ANY ANGLE, RADIUS BEING = 1·00.

Degrees.					
°	Arc.	°	Arc.	°	Arc.
1	·0174533	31	·5410521	61	1·0646508
2	·0349066	32	·5585054	62	1·0821041
3	·0523599	33	·5759587	63	1·0995574
4	·0698132	34	·5934119	64	1·1170107
5	·0872665	35	·6108652	65	1·1344640
6	·1047198	36	·6283185	66	1·1519173
7	·1221730	37	·6457718	67	1·1693706
8	·1396263	38	·6632251	68	1·1868239
9	·1570796	39	·6806784	69	1·2042772
10	·1745329	40	·6981317	70	1·2217305
11	·1919862	41	·7155850	71	1·2391828
12	·2094395	42	·7330383	72	1·2566371
13	·2268928	43	·7504916	73	1·2740904
14	·2443461	44	·7679449	74	1·2915436
15	·2617994	45	·7853982	75	1·3089969
16	·2792527	46	·8028515	76	1·3264502
17	·2967060	47	·8203047	77	1·3439035
18	·3141593	48	·8377580	78	1·3613568
19	·3316126	49	·8552113	79	1·3788101
20	·3490659	50	·8726646	80	1·3962634
21	·3665191	51	·8901179	81	1·4137167
22	·3839724	52	·9075712	82	1·4311700
23	·4014257	53	·9250245	83	1·4486233
24	·4188790	54	·9424778	84	1·4660766
25	·4363323	55	·9599311	85	1·4835299
26	·4537856	56	·9773844	86	1·5009832
27	·4712389	57	·9948377	87	1·5184364
28	·4886922	58	1·0122910	88	1·5358897
29	·5061455	59	1·0297443	89	1·5533430
30	·5235988	60	1·0471976	90	1·5707963

CONVERSION OF INCHES AND EIGHTHS INTO DECIMALS OF A FOOT.

Inches.	Fractions of an Inch.							
	0	⅛	¼	⅜	½	⅝	¾	⅞
0	·0000	·01041	·02083	·03125	·04166	·05208	·0625	·07291
1	·08333	·09375	·10416	·11458	·125	·13541	·14588	·15639
2	·16666	·17707	·1875	·19792	·20832	·21873	·22914	·23965
3	·25	·26041	·27	·28125	·29166	·30208	·3125	·32291
4	·33333	·34375	·35416	·364	·375	·38541	·39588	·40639
5	·41666	·42707	·437	·44792	·45832	·46873	·47914	·48965
6	·5	·51041	·520	·53125	·54166	·55208	·5625	·57291
7	·58333	·59375	·60416	·614	·625	·63541	·64588	·65639
8	·66666	·67707	·685	·69792	·70832	·71773	·72914	·73965
9	·75	·76041	·77	·78125	·79169	·80208	·8125	·82291
10	·83333	·84375	·85416	·864	·875	·88541	·89588	·90639
11	·91666	·92707	·937	·94792	·95832	·96873	·97914	·98965
12	1 foot.	foot.	foot.	foot.	foot.	foot.	foot.	foot.

DECIMAL EQUIVALENTS OF A TON.

cwt.	qr.	lb.	tons.	cwt.	qr.	lb.	tons.	cwt.	qr.	lb.	tons.	cwt.	qr.	lb.	tons.	cwt.	qr.	lb.	tons.
0	0	22½ =	·01	4	0	22½ =	·21	8	0	22½ =	·41	12	0	22½ =	·61	16	0	22½ =	·81
0	1	16¾ =	·02	4	1	16¾ =	·22	8	1	16¾ =	·42	12	1	16¾ =	·62	16	1	16¾ =	·82
0	2	11¼ =	·03	4	2	11¼ =	·23	8	2	11¼ =	·43	12	2	11¼ =	·63	16	2	11¼ =	·83
0	3	5½ =	·04	4	3	5½ =	·24	8	3	5½ =	·44	12	3	5½ =	·64	16	3	5½ =	·84
1	0	0 =	·05	5	0	0 =	·25	9	0	0 =	·45	13	0	0 =	·65	17	0	0 =	·85
1	0	22½ =	·06	5	0	22½ =	·26	9	0	22½ =	·46	13	0	22½ =	·66	17	0	22½ =	·86
1	1	16¾ =	·07	5	1	16¾ =	·27	9	1	16¾ =	·47	13	1	16¾ =	·67	17	1	16¾ =	·87
1	2	11¼ =	·08	5	2	11¼ =	·28	9	2	11¼ =	·48	13	2	11¼ =	·68	17	2	11¼ =	·88
1	3	5½ =	·09	5	3	5½ =	·29	9	3	5½ =	·49	13	3	5½ =	·69	17	3	5½ =	·89
2	0	0 =	·1	6	0	0 =	·3	10	0	0 =	·5	14	0	0 =	·7	18	0	0 =	·9
2	0	22½ =	·11	6	0	22½ =	·31	10	0	22½ =	·51	14	0	22½ =	·71	18	0	22½ =	·91
2	1	16¾ =	·12	6	1	16¾ =	·32	10	1	16¾ =	·52	14	1	16¾ =	·72	18	1	16¾ =	·92
2	2	11¼ =	·13	6	2	11¼ =	·33	10	2	11¼ =	·53	14	2	11¼ =	·73	18	2	11¼ =	·93
2	3	5½ =	·14	6	3	5½ =	·34	10	3	5½ =	·54	14	3	5½ =	·74	18	3	5½ =	·94
3	0	0 =	·15	7	0	0 =	·35	11	0	0 =	·55	15	0	0 =	·75	19	0	0 =	·95
3	0	22½ =	·16	7	0	22½ =	·36	11	0	22½ =	·56	15	0	22½ =	·76	19	0	22½ =	·96
3	1	16¾ =	·17	7	1	16¾ =	·37	11	1	16¾ =	·57	15	1	16¾ =	·77	19	1	16¾ =	·97
3	2	11¼ =	·18	7	2	11¼ =	·38	11	2	11¼ =	·58	15	2	11¼ =	·78	19	2	11¼ =	·98
3	3	5½ =	·19	7	3	5½ =	·39	11	3	5½ =	·59	15	3	5½ =	·79	19	3	5½ =	·99
4	0	0 =	·2	8	0	0 =	·4	12	0	0 =	·6	16	0	0 =	·8	20	0	0 =	1·0

WEIGHT AND BULK OF SUBSTANCES.

Names of Substances.	Weight of Cubic foot in pounds.	Cubic feet in ton.	Names of Substances.	Weight of Cubic foot in pounds.	Cubic feet in ton.
Cast iron	450·5	4·97	Mahogany	66·4	33·8
Wrought iron	486·6	4·60	Oak, live (American)	70	32
Steel	489·8	4·57	„ white „	45·2	49·5
Copper	555	4·03	„ „ (English)	53	42
Lead	707·7	3·16	Cork	15	150
Brass	537·7	4·16	Coal	79	28·3*
Tin	456	4·91	Coke	46	48·7
Gold	1013	2·21	Marble, common	141	15·9
Silver	551	4·07	Millstone	130	17·2
Pine, white (spruce fir)	29·56	75·6	Clay	101·3	22·1
„ yellow	33·81	66·2	Sand	94·5	23·7
„ red (Riga or Dantzic)	40	55	Granite	165	13·5
„ pitch	41·3	54	Earth, loose	78·6	28·5
English elm	35	64	Water, salt (sea)	64·3	34·8
American elm	45	50	„ fresh	62·5	35·9
Teak	50	44	Ice	58·08	38·56

* This assumes the coal to be in one solid block. For broken coal 40 cubic feet are allowed for stowage in bunkers.

KNOTS PER HOUR CONVERTED INTO FEET PER SECOND.

Knots per hour.	Feet per second.	Knots per hour.	Feet per second.	Knots per hour.	Feet per second.	Knots per hour.	Feet per second.	Knots per hour.	Feet per second.
1	1·688	7	11·82	13	21·94	19	32·07	25	42·20
2	3·376	8	13·50	14	23·63	20	33·76	26	43·89
3	5·064	9	15·19	15	25·32	21	35·45	27	45·58
4	6·752	10	16·88	16	27·01	22	37·14	28	47·26
5	8·44	11	18·57	17	28·70	23	38·82	27	48·95
6	10·13	12	20·26	18	30·38	24	40·51	30	50·64

The Admiralty knot is 6080 feet. A statute mile is = knot × ·8684. A knot is = statute mile × 1·1515.

TIME AND KNOT TABLE.

To use this table note the time on the mile, then find the number of seconds in the first column and trace along horizontally until the column for minutes is reached; the figures in that column will be the rate in knots per hour. Say the time was 4 min. 29 sec. Find 29 seconds, and trace along to the 4 minute column; the figures are 13·383, which is the rate per hour.

Secs.	2 min.	3 min.	4 min.	5 min.	6 min.	7 min.	8 min.	9 min.	10 min.	11 min.	12 min.	13 min.	14 min.
0	30·000	20·000	15·000	12·000	10·000	8·571	7·500	6·667	6·000	5·455	5·000	4·615	4·286
1	29·752	19·890	14·938	11·960	9·972	8·551	7·484	6·654	5·990	5·446	4·993	4·609	4·281
2	29·508	19·780	14·876	11·921	9·945	8·531	7·469	6·642	5·980	5·438	4·986	4·604	4·275
3	29·268	19·672	14·815	11·881	9·917	8·511	7·453	6·630	5·970	5·430	4·979	4·598	4·270
4	29·032	19·565	14·754	11·842	9·890	8·491	7·438	6·618	5·960	5·422	4·972	4·592	4·265
5	28·800	19·459	14·694	11·803	9·863	8·471	7·423	6·606	5·950	5·414	4·965	4·586	4·260
6	28·571	19·355	14·634	11·765	9·836	8·451	7·407	6·593	5·941	5·405	4·959	4·580	4·255
7	28·346	19·251	14·575	11·726	9·809	8·431	7·392	6·581	5·931	5·397	4·952	4·574	4·250
8	28·125	19·149	14·516	11·688	9·783	8·411	7·377	6·569	5·921	5·389	4·945	4·568	4·245
9	27·907	19·048	14·458	11·650	9·756	8·392	7·362	6·557	5·911	5·381	4·938	4·563	4·240
10	27·692	18·947	14·400	11·613	9·730	8·372	7·347	6·545	5·902	5·373	4·931	4·557	4·235
11	27·481	18·848	14·343	11·576	9·704	8·353	7·332	6·534	5·892	5·365	4·925	4·551	4·230
12	27·273	18·750	14·286	11·538	9·677	8·333	7·317	6·522	5·882	5·357	4·918	4·545	4·225
13	27·068	18·653	14·229	11·502	9·651	8·314	7·302	6·510	5·873	5·349	4·911	4·540	4·220
14	26·866	18·557	14·173	11·465	9·626	8·295	7·287	6·498	5·863	5·341	4·905	4·534	4·215
15	26·667	18·461	14·118	11·429	9·600	8·276	7·273	6·486	5·854	5·333	4·898	4·528	4·210
16	26·471	18·367	14·062	11·392	9·574	8·257	7·258	6·475	5·844	5·325	4·891	4·523	4·206
17	26·277	18·274	14·008	11·356	9·549	8·238	7·243	6·463	5·835	5·318	4·885	4·517	4·201
18	26·087	18·182	13·953	11·321	9·524	8·219	7·229	6·452	5·825	5·310	4·878	4·511	4·196
19	25·899	18·090	13·900	11·285	9·499	8·200	7·214	6·440	5·816	5 302	4·871	4·506	4·191
20	25·714	18·000	13·846	11·250	9·474	8·182	7·200	6·429	5·806	5·294	4·865	4·500	4·186
21	25·532	17·910	13·793	11·215	9·449	8·163	7·186	6·417	5·797	5·286	4·858	4·494	4·181
22	25·352	17·822	13·740	11·180	9·424	8·145	7·171	6·406	5·788	5·279	4·852	4·489	4·176
23	25·175	17·734	13·688	11·146	9·399	8·126	7·157	6·394	5·778	5·271	4·845	4·483	4·171
24	25·000	17·647	13·636	11·111	9·375	8·108	7·143	6·383	5·769	5·263	4·839	4·478	4·167
25	24·828	17·561	13·585	11·077	9·351	8·090	7·129	6·372	5·760	5·255	4·832	4·472	4·162
26	24·658	17·476	13·534	11·043	9·326	8·072	7·115	6·360	5·751	5·248	4·826	4·466	4·157
27	24·490	17·391	13·483	11·009	9·302	8·054	7·101	6·349	5·742	5·240	4·819	4·461	4·152
28	24·324	17·308	13·433	10·976	9·278	8·036	7·087	6·338	5·732	5·233	4·813	4·455	4·147
29	24·161	17·225	13·383	10·942	9·254	8·018	7·073	6·327	5·723	5·225	4·806	4·450	4·143
30	24·000	17·143	13·333	10·909	9·231	8·000	7·059	6·316	5·714	5·217	4·800	4·444	4·138
31	23·841	17·062	13·284	10·876	9·207	7·982	7·045	6 305	5·705	5·210	4·794	4·439	4·133
32	23·684	16·981	13·235	10·843	9·184	7·965	7·031	6·294	5·696	5·202	4·787	4·433	4·128
33	23·529	16·901	13·187	10·811	9·160	7·947	7·018	6·283	5·687	5·195	4·781	4·428	4·124
34	23·377	16·822	13·139	10·778	9·137	7·930	7·004	6·272	5·678	5·187	4·774	4·423	4·119
35	23·226	16·744	13·091	10·746	9·114	7·912	6·990	6·261	5·669	5·180	4·768	4·417	4·114
36	23·077	16·667	13·043	10·714	9·091	7·895	6·977	6·250	5·660	5·172	4·762	4·412	4·110
37	22·930	16·590	12·996	10·682	9·068	7·877	6·963	6·239	5·651	5·165	4·756	4·406	4·105
38	22·785	16·514	12·950	10·651	9·045	7·860	6·950	6·228	5·643	5·158	4·749	4·401	4·100
39	22·642	16·438	12·903	10·619	9·023	7·843	9·936	6·218	5·684	5·150	4·743	4·396	4·096
40	22·500	16·364	12·857	10·588	9·000	7·826	6·923	6·207	5·625	5·143	4·737	4·390	4·091
41	22·360	16·290	12·811	10·557	8·978	7·809	6·910	6·196	5·616	5·136	4·731	4·385	4·086
42	22·222	16·216	12·766	10·526	8·955	7·792	6·897	6·186	5·607	5·128	4·724	4·379	4·082
43	22·086	16·143	12·721	10·496	8·933	7·775	6·883	6·175	5·599	5·121	4·718	4·374	4·077
44	21·951	16·071	12·676	10·465	8·911	7·759	6·870	6·164	5·590	5·114	4·712	4·369	4·072
45	21·818	16·000	12·632	10·435	8·889	7·742	6·857	6·154	5·581	5·106	4·706	4·364	4·068
46	21·687	15·929	12·587	10·405	8·867	7·725	6·844	6·143	5·573	5·099	4·700	4·358	4·063
47	21·557	15·859	12·544	10·375	8·845	7·709	6·831	6·133	5·564	5·092	4·693	4·353	4·059
48	21·429	15·789	12·500	10·345	8·824	7·692	6·818	6·122	5·556	5·085	4·687	4·348	4·054
49	21·302	15·721	12·457	10·315	8·802	7·676	6·805	6·112	5·547	5·078	4·681	4·343	4·049
50	21·176	15·652	12·414	10·286	8·780	7·660	6·792	6·102	5·538	5·070	4·675	4·337	4·045
51	21·053	15·584	12·371	10·256	8·759	7·643	6·780	6·091	5·530	5·063	4·669	4·332	4·040
52	20·930	15·517	12·329	10·227	8·738	7·627	6·767	6·081	5·521	5·056	4·663	4·327	4·035
53	20·809	15·451	12·287	10·198	8·717	7·611	6·754	6·071	5·513	5·049	4·657	4·322	4·031
54	20·690	15·385	12·245	10·169	8·696	7·595	6·742	6·061	5·505	5·042	4·651	4·316	4·027
55	20·571	15·319	12·203	10·141	8·675	7·579	6·729	6·050	5·496	5·035	4·645	4·311	4·022
56	20·455	15·254	12·162	10·112	8·654	7·563	6·716	6·040	5·488	5·028	4·639	4·306	4·018
57	20·339	15·190	12·121	10·084	8·633	7·547	6·704	6 030	5·479	5·021	4·633	4·301	4·013
58	20·225	15·126	12·081	10·056	8·612	7·531	6·691	6·020	5·471	5·014	4·627	4·296	4·009
59	20·112	15·063	12·040	10·028	8·592	7·516	6·679	6·010	5·463	5·007	4·621	4·291	4·004
Secs.	2 min.	3 min.	4 min.	5 min.	6 min.	7 min.	8 min.	9 min.	10 min.	11 min.	12 min.	13 min.	14 min.

WEIGHT AND STRENGTH OF HEMP AND WIRE ROPE.

Hemp.		Iron Wire.		Steel Wire.		Equivalent Strength for the Circumferences given.	
Circumference.	Pounds weight per Fathom.	Circumference for equivalent Strength.	Pounds weight per Fathom.	Circumference for Equivalent Strength.	Pounds weight per Fathom.	Working Load in Cwts.	Breaking strain in Tons.
2¾	2	1	1	...	...	6	2
...	...	1½	1½	1	1	9	3
3¾	4	1⅝	2	...	...	12	4
...	...	1¾	2½	1½	1½	15	5
4½	5	1⅞	3	...	...	18	6
...	...	2	3½	1⅝	2	21	7
5½	7	2⅛	4	1¾	2½	24	8
...	...	2¼	4½	...	...	27	9
6	9	2⅜	5	1⅞	3	30	10
...	...	2½	5½	...	...	33	11
6½	10	2⅝	6	2	3½	36	12
...	...	2¾	6½	2⅛	4	39	13
7	12	2⅞	7	2¼	4½	42	14
...	...	3	7½	...	...	45	15
7½	14	3⅛	8	2⅜	5	48	16
...	...	3¼	8½	...	...	51	17
8	16	3⅜	9	2½	5½	54	18
...	...	3½	19	2⅝	6	60	20
8½	18	3⅝	11	2¾	6½	66	22
...	...	3¾	12	...	...	72	24
9½	22	3⅞	13	3¼	8	78	26
10	25	4	14	...	...	84	28
...	...	4¼	15	3⅜	9	90	30
11	30	4⅜	16	...	...	96	32
...	...	4½	18	3½	10	08	36
12	36	4⅝	20	3¾	12	120	40

Manilla rope, if not dried up and chafed, is slightly stronger size for size than hemp.

WEIGHT OF A SUPER. FOOT OF PLATES, OF DIFFERENT METALS, IN LB.

Thickness.	Iron.	Steel.	Brass.	Copper.	Lead.	Zinc.
in.	lb.	lb.	lb.	lb.	lb.	lb.
$\frac{1}{16}$	2·5	2·6	2·7	2·9	3·7	2·3
$\frac{1}{8}$	5·0	5·2	5·5	5·8	7·4	4·7
$\frac{3}{16}$	7·5	7·8	8·2	8·7	11·1	7·0
$\frac{1}{4}$	10·0	10·4	11·0	11·6	14·8	9·4
$\frac{5}{16}$	12·5	13·0	13·7	14·5	18·5	11·7
$\frac{3}{8}$	15·0	15·6	16·4	17·2	22·2	14·0
$\frac{7}{16}$	17·5	18·2	19·2	20·0	25·9	16·4
$\frac{1}{2}$	20·0	20·8	21·9	22·9	29·5	18·7
$\frac{9}{16}$	22·5	23·4	24·6	25·8	33·2	21·1
$\frac{5}{8}$	25·0	26·0	27·4	28·6	36·9	23·4
$\frac{11}{16}$	27·5	28·6	30·1	31·4	40·6	25·7
$\frac{3}{4}$	30·0	31·2	32·9	34·3	44·3	28·1
$\frac{13}{16}$	32·5	33·8	35·6	37·2	48·0	30·4
$\frac{7}{8}$	35·0	36·4	38·3	40·0	51·7	32·8
$\frac{15}{16}$	37·5	39·0	41·2	42·9	55·4	35·1
1	40·0	41·6	43·9	45·8	59·1	37·5

WEIGHT OF CHAINS.

Chains.		Chain Cables.	
Diameter in inches.	Weight per Fathom in lb.	Diameter in inches.	Weight per Fathom in lb.
$\frac{5}{16}$	5¼	½	13¾
⅜	8	⅝	22
½	14	¾	30
⅝	22	⅞	42
¾	32	1	55
⅞	43	1⅛	68
1	56	1¼	84
1⅛	71	1⅜	102
1¼	87	1½	120
1⅜	106	1¾	148
		2	180

TABLE OF 1·8 ROOTS (*see* PAGE 17).

(*see* PAGE 17)

The 1·8 power of the roots will of course be the numbers in the first column.

Numbers.	1·8 Roots of Numbers.	Numbers.	1·8 Roots of Numbers.	Numbers.	1·8 Roots of Numbers.	Numbers.	1·8 Roots of Numbers.
1·0	1·000	7·0	2·950	40	7·77	100	12·91
1·1	1·060	7·1	2·973	41	7·88	101	12·98
1·2	1·115	7·2	2·996	42	7·99	102	13·05
1·3	1·168	7·3	3·018	43	8·09	103	13·12
1·4	1·218	7·4	3·040	44	8·19	104	13·19
1·5	1·265	7·5	3·062	45	8·29	105	13·26
1·6	1·310	7·6	3·084	46	8·39	106	13·33
1·7	1·352	7·7	3·106	47	8·49	107	13·40
1·8	1·392	7·8	3·128	48	8·59	108	13·47
1·9	1·432	7·9	3·150	49	8·69	109	13·54
2·0	1·471	8·0	3·172	50	8·79	110	13·61
2·1	1·510	8·1	3·194	51	8·89	111	13·68
2·2	1·549	8·2	3·216	52	8·99	112	13·75
2·3	1·587	8·3	3·237	53	9·08	113	13·82
2·4	1·625	8·4	3·259	54	9·17	114	13·89
2·5	1·662	8·5	3·281	55	9·26	115	13·96
2·6	1·698	8·6	3·303	56	9·35	116	14·03
2·7	1·734	8·7	3·325	57	9·44	117	14·10
2·8	1·770	8·8	3·347	58	9·53	118	14·17
2·9	1·805	8·9	3·369	59	9·62	119	14·24
3·0	1·840	9·0	3·390	60	9·71	120	14·31
3·1	1·874	9·1	3·410	61	9·80	121	14·38
3·2	1·907	9·2	3·430	62	9·89	122	14·45
3·3	1·940	9·3	3·450	63	9·98	123	14·52
3·4	1·973	9·4	3·470	64	10·07	124	14·59
3·5	2·005	9·5	3·490	65	10·16	125	14·66
3·6	2·037	9·6	3·510	66	10·25	126	14·73
3·7	2·069	9·7	3·530	67	10·34	127	14·80
3·8	2·100	9·8	3·550	68	10·43	128	14·86
3·9	2·130	9·9	3·570	69	10·52	129	14·92
4·0	2·160	10	3·590	70	10·60	130	14·98
4·1	2·190	11	3·790	71	10·68	131	15·04
4·2	2·220	12	3·98	72	10·76	132	15·10
4·3	2·250	13	4·16	73	10·84	133	15·16
4·4	2·280	14	4·33	74	10·92	134	15·22
4·5	2·309	15	4·50	75	11·00	135	15·28
4·6	2·338	16	4·66	76	11·08	136	15·34
4·7	2·366	17	4·82	77	11·16	137	15·40
4·8	2·394	18	4·97	78	11·24	138	15·46
4·9	2·422	19	5·12	79	11·32	139	15·52
5·0	2·450	20	5·27	80	11·40	140	15·58
5·1	2·477	21	5·42	81	11·48	141	15·64
5·2	2·504	22	5·56	82	11·56	142	15·70
5·3	2·531	23	5·70	83	11·64	143	15·76
5·4	2·557	24	5·84	84	11·72	144	15·82
5·5	2·583	25	5·97	85	11·80	145	15·88
5·6	2·609	26	6·10	86	11·88	146	15·94
5·7	2·635	27	6·23	87	11·96	147	16·00
5·8	2·660	28	6·36	88	12·04	148	16·06
5·9	2·685	29	6·49	89	12·12	149	16·12
6·0	2·710	30	6·62	90	12·20	150	16·18
6·1	2·735	31	6·74	91	12·28	151	16·24
6·2	2·760	32	6·86	92	12·35	152	16·30
6·3	2·784	33	6·98	93	12·42	153	16·36
6·4	2·808	34	7·10	94	12·49	154	16·42
6·5	2·832	35	7·22	95	12·56	155	16·48
6·6	2·856	36	7·33	96	12·63	156	16·54
6·7	2·880	37	7·44	97	12·70	157	16·60
6·8	2·904	38	7·55	98	12·77	158	16·66
6·9	2·927	39	7·66	99	12·84	159	16·72

TABLE OF 1·8 ROOTS (*continued*).

Numbers.	1·8 Roots of Numbers.	Numbers.	1·8 Roots of Numbers.	Numbers.	1·8 Roots of Numbers.	Numbers.	1·8 Roots of Numbers.
160	16·78	225	20·25	289	23·42	353	25·98
161	16·84	226	20·30	290	23·46	354	26·02
162	16·90	227	20·35	291	23·50	355	26·06
163	16·96	228	20·40	292	23·54	356	26·10
164	17·02	229	20·45	293	23·58	357	26·14
165	17·08	230	20·50	294	23·62	358	26·18
166	17·14	231	20·55	295	23·66	359	26·22
167	17·20	232	20·60	296	23·70	360	26·26
168	17·26	233	20·65	297	23·74	361	26·30
169	17·32	234	20·70	298	23·78	362	26·34
170	17·38	235	20·75	299	23·82	363	26·38
171	17·44	236	20·80	300	23·86	364	26·42
172	17·50	237	20·85	301	23·90	365	26·46
173	17·56	238	20·90	302	23·94	366	26·50
174	17·62	239	20·95	303	23·98	367	26·54
175	17·68	240	21·00	304	24·02	368	26·58
176	17·74	241	21·05	305	24·06	369	26·62
177	17·80	242	21·10	306	24·10	370	26·66
178	17·86	243	21·15	307	24·14	371	26·70
179	17·92	244	21·20	308	24·18	372	26·74
180	17·98	245	21·25	309	24·22	373	26·78
181	18·04	246	21·30	310	24·26	374	26·82
182	18·10	247	21·35	311	24·30	375	26·86
183	18·15	248	21·40	312	24·34	376	26·90
184	18·20	249	21·45	313	24·38	377	26·94
185	18·25	250	21·50	314	24·42	378	26·98
186	18·30	221	21·55	315	24·46	379	27·02
187	18·35	252	21·60	316	24·50	380	27·06
188	18·40	253	21·65	317	24·54	381	27·10
189	18·45	254	21·70	318	24·58	382	27·14
190	18·50	255	21·75	319	24·62	383	27·18
191	18·55	256	21·80	320	24·66	384	27·22
192	18·60	257	21·85	321	24·70	385	27·26
193	18·65	258	21·90	322	24·74	386	27·30
194	18·70	259	21·95	323	24·78	387	27·34
195	18·75	260	22·00	324	24·82	388	27·38
196	18·80	261	22·05	325	24·86	389	27·42
197	18·85	262	22·10	326	24·90	390	27·46
198	18·90	263	22·15	327	24·94	391	27·50
199	18·95	264	22·20	328	24·98	392	27·54
200	19·00	265	22·25	329	25·02	393	27·58
201	19·05	266	22·30	330	25·06	394	27·62
202	19·10	267	22·35	331	25·10	395	27·66
203	19·15	268	22·40	332	25·14	396	27·70
204	19·20	269	22·45	333	25·18	397	27·74
205	19·25	270	22·50	334	25·22	398	27·78
206	19·30	271	22·55	335	25·26	399	27·81
207	19·35	272	22·60	336	25·30	400	27·84
208	19·40	273	22·65	337	25·34	412	28·71
209	19·45	274	22·70	338	25·38	455	30·00
210	19·50	275	22·75	339	25·42	500	31·61
211	19·55	276	22·80	340	25·46	560	33·63
212	19·60	277	22·85	341	25·50	630	35·93
213	19·65	278	22·90	342	25·54	652	36·40
214	19·70	279	22·95	343	25·58	700	38·07
215	19·75	280	23·00	344	25·62	790	40·80
216	19·80	281	23·05	345	25·66	800	41·00
217	19·85	282	23·10	346	25·70	900	43·78
218	19·90	283	23·15	347	25·74	993	46·24
219	19·95	284	23·20	348	25·78	1500	58·15
220	20·00	285	23·25	349	25·82	2000	68·22
221	20·05	286	23·30	350	25·86	2372	75·00
222	20·10	287	23·34	351	25·90	3426	92·00
223	20·15	288	23·38	352	25·94	3982	100·00
224	20·20						

MARINE GLUE.

This composition is said to be composed of 1 part india-rubber, 12 mineral naphtha or coal tar heated gently, and 20 parts of shellac, mixed with it. The composition is now usually employed to stop the seams of decks after they are caulked. The old fashioned plan was to use white lead putty for the stopping, and indeed it is at this present time occasionally used; the objection to it is that it dries as hard as a cement and cracks, the result being that water gets into the caulking, rots it, and then leaky decks are the consequence. Moreover, hard putty is very difficult to get out of the seams without damaging the edges of the plank, and then in re-stopping ragged ugly seams are the result. Marine glue, on the other hand, if it does dry and crack, can easily be renewed, and the edges of the plank remain uninjured.

In using marine glue the following practice should be observed: In driving the oakum or cotton thread (the latter is sometimes preferred as it can be laid in finer strands, a matter of consideration if the plank is closely laid) into the seams, the caulking iron should be dipped in naphtha, and not in oil, as, if the sides of the plank are touched with the latter, the glue will not adhere; naphtha on the other hand dissolves the glue and assists in closely cementing the seams. The plank should be quite dry when the glue is applied, or it will not adhere to the sides of the seams. The glue should be dissolved in a pot, and applied by lip ladles used for paying, two being kept going; or the glue can be melted in the lip ladles. Great care must be taken that the glue is melted slowly, as if it be melted over too fierce a fire it will be spoilt. A little of the liquid glue can be usefully mixed with the other, as it assists in keeping it dissolved. The glue that runs over the sides of the seams should be cleaned off with a broad sharp chisel and remelted. It is not advisable to scrape the surplus glue off the seams, as it cannot be so removed without leaving a ragged unsightly surface. The manufacturer of this marine glue is Mr. Jeffery, Limehouse.

A cheaper kind of marine glue is now much used. It is manufactured by the Landport Waterproof Company, and is much more easy to use than the other, as it does not lose its qualities by heating.

YACHT ARCHITECTURE IN GERMANY.

THE continued and rapid growth of yachting in Germany, under the fostering care of the Emperor and his brother, Prince Henry of Prussia, has had the immediate result of giving a fresh impetus to the science of designing and construction.

In a country where the sport of yacht racing is as yet in its infancy, it is only natural that a commencement should be made with the building of small boats. Heidtmann's yard on the Alster, at Hamburg, for example, has turned out some first-rate boats, capable of competing in all respects with the average small racer of British build;* but so far the demand for large yachts has been so small that German builders have scarcely had an opportunity of gaining experience in their construction.

Not only Great Britain, but America as well, has had some influence on German yacht architecture. A 1-rater, Dr. von Türk's Bubble, and the late Baron von Zedtwitz's 20-rater Isolde (a sister ship of Mr. Howard Gould's Niagara), are the only direct importations of recent years. Prince Henry of Prussia's Gudruda, the Herreshoff crack Wenonah, was for some time raced on the Clyde by Mr. H. Allan before being sold to Germany.

It may be said that sixteen years ago, not only yacht architecture, but the sport of yachting itself, was practically unknown in Germany. There were owned at Hamburg a few cruising craft of British origin; but there was no racing, unless we count that of small boats on the Alster, and on the network of lakes that surround Berlin. In 1882, however, some German naval officers arranged a regatta on the fjord of Kiel, which proved such a success that the Norddeutscher Regatta Verein took up the idea, and offered prizes for annual competition on the Baltic. This attracted a good many foreign yachts, especially from Denmark and Sweden; and five years later was formed the Marine Regatta Verein, out of which has developed the Imperial Yacht Club. The history of these two clubs is the history of yachting in Germany.

The Norddeutscher Regatta Verein was founded in 1868, primarily for rowing and sailing regattas on the Alster, Hamburg's picturesque lake. Later on it established fixtures on the Elbe, which remain to this day and ultimately the matches on the Baltic at Kiel, and in the bay of Lübeck

* The principal building yard on the Alster, established in 1855, has shown a steady increase in its output. Of the seventy-five craft launched from Mr. Heidtmann's yard in 1894, thirteen were *bonâ fide* yachts; in 1895, twelve out of eighty-one were sailing yachts; while in 1896, previous to the Kiel Regattas, eighteen sailing yachts had been despatched to their various destinations.

at Travemünde, as the principal field of operations. The Empress Frederick is the club's patron, and its only honorary members are H.R.H. Prince Henry of Prussia and the hereditary Grand Duke Frederick Augustus of Oldenburg.

The Commodore of the Imperial Yacht Club is the German Emperor, and Prince Henry is vice-commodore. The rapid growth of the club is due in great measure to the energy of the secretary, Geheimer Regierungsrath Professor Busley, until recently Professor of Naval Construction at the Kaiserliche Marine Akademie at Kiel, now managing director of Schichau's yard at Elbing, who for many years has acted as timekeeper and judge at the annual regattas.

The Baltic is an ideal yachting ground—for cruising as well as for racing. Practically, there are no tides to contend with. Germany's great naval station, Kiel, therefore, has unusually favourable chances of becoming a great yachting centre. Its large and sheltered harbour, the wide fjord, and the open Baltic beyond, with plenty of ports for which to run in case of dirty weather, can hardly be surpassed. Another great advantage it possesses as a yachting centre, which has not been without its beneficial influence on the development of yacht architecture in Germany, is its proximity to the Danish islands and to Sweden, in both of which countries the sport is strongly developed. With the exception of 1894, when there was a temporary breach owing to some political misunderstanding, caused by the expulsion of a troupe of Danish actors from Hadersleben, Danish and Swedish yachts, well built, beautifully kept, and skilfully handled have taken part in regattas at Kiel for many years, and have done much to stimulate German owners. It is from Denmark, too, that the formula comes under which German racing yachts are measured and rated—Mr. Alfred Benzon, a member of the Royal Danish Yacht Club, being domiciled at Copenhagen.* The formula, which has been adopted by the Deutscher Segler Verband (the German equivalent for our own Y.R.A.), and which cannot be altered until 1898, is as follows:

$$R = \frac{L \times G\ (L + \sqrt{S})}{150} \text{ “Sail-units.”}$$

L signifies the length of the yacht on the water-line in metres.

G signifies the girth of the yacht in metres.

S signifies the sail area of the yacht in square metres.

* The effect of the Benzon rule appears to have been the revival of the old style of boat which was in use at Hamburg, on the Alster, in the sixties. Thus, in 1864, Mr. A. Tietgens purchased the Laura, built in America three years before, introducing a type which, after three decades, has been resuscitated. The Schelm, built by Heidtmann for Mr. A. Kirsten in 1865, save for heavier build and greater strength, was almost an exact counterpart of the modern German small rater.

The length on the water-line is found by deducting the overhangs from the length over all, one or two special cases being provided for. Notably in the case of extravagant overhangs the following rule has to be taken into account:

"Should the length over all (L_1) exceed the length on the L.W.L. by more than 50 per cent, *i.e.* (L_1 more than 1·5 L.W.L.), such greater length ($L_2 = L_1 - 1{\cdot}5$ L.W.L) is added to the L.W.L. (L.W.L. + L_2), and taxed accordingly."

The girth is obtained by the following formula:

$$G = P + \left(\frac{B + B_1}{2}\right)$$

Here the greatest under-water girth is taken by a chain up to the water-line. Freeboard is not taxed. To this under-water girth P, the arithmetical mean of $B + B_1$ is added. B in this case signifies the maximum beam on L.W.L., and B_1 the extreme beam wherever found.

Centre-board yachts, built after January 1, 1894, are dealt with in the following manner: If their draught with housed centre-board does not exceed 1·5 metres, the girth is taken with housed centre-board (G_1), and with the centre-plate down (G), and the factor G is then determined by this formula:

$$G = G_1 + \left(\frac{G_2 - G_1}{3}\right)$$

The factor G_2 is taken with the centre-plate as far down as the owner states he is in the habit of ever using it. In the case of yachts above 1·5 metres draught (of hull), the girth G is taken with centre-board down.

The sail area is measured in exactly the same manner as prescribed by the Y.R.A. rules.

It was the Emperor's purchase on the Clyde of the Thistle, the unsuccessful challenger for the America Cup in 1887, that gave yachting in Germany its first great impetus. At the same time, Prince Henry of Prussia placed an order with Mr. G. L. Watson for the construction of a cruiser-racer of 40-rating, which resulted in the Clyde-built Irene, so called after the owner's consort. For a few seasons this yacht had matters all her own way on the Baltic, but in 1894, when Varuna and Lais hoisted their racing flags for the first time in German waters, when the Mücke made her *début,* and Admiral the Hon. Victor Montagu started his 40-rater Carina, it became evident that the Irene was no match for these more modern flyers. In consequence, she was not entered for any matches in 1896, and replaced in the following year by Lord Dunraven's L'Esperance.

The boat *par excellence* which has had the greatest influence on German yacht architecture is Prince Henry of Prussia's Gudruda (*née* Wenonah). By special permission granted by her Royal owner an exact replica was built for Mr. Bichel, which, under the name of Swanhild, has proved a formidable rival to Prince Henry's boat.

In the spring of 1895 the German Emperor called to life a 20-rating class by the construction, at Kiel, of the Vineta, from designs by Mr. G. L. Watson. This craft, which under the old Y.R.A. measurement ranks as a 20-rater, and according to the Benzon rule is of 32 "sail-units," is of composite build. Unfortunately, owing, it may be, to too light a construction, the yacht, in one of her first races, was severely strained in the short, choppy seas of the "Stollergrund," a bank at the mouth of the fjord of Kiel, and no subsequent patching seemed to do her any good. In the same year the Vineta was entered and raced most pluckily in the regattas on the Solent, but without success. On the advent of the second Meteor, the Emperor presented the Vineta to his brother-in-law, Prince Ferdinand of Schleswig-Holstein-Glücksburg, and under his flag she was entered at the regattas on the Baltic in 1896; but against so formidable an opponent as the late Baron von Zedtwitz's Isolde, to which, moreover, she had to concede time, she never had the ghost of a chance.

The Hertha, Ellen, and Elizabeth, each of 30·5 "sail-units," were also built to compete with the Emperor's Vineta. The first and last mentioned were constructed in Germany, the Elizabeth from designs by Mr. W. Fife, jun., while the Ellen was built at Gourock, on the Clyde, from designs by Mr. G. L. Watson.

For the further encouragement of yacht architecture in Germany the Emperor offered an annual money prize, varying with the size of the successful vessel, for a craft from the board of a German designer, built within Germany with German material, and manned by a German crew. The first practical outcome was the formation of a syndicate of members of the Imperial Yacht Club at Kiel, who built the Kommodore (the design of which appears elsewhere), in accordance with these conditions, and won the prize. At the end of her first season the Kommodore was sold to the Grand Duke of Mecklenburg-Schwerin, and a second yacht was laid down by the syndicate, the Hertha, which, however, did not prove quite so successful as the Kommodore.

It is certainly worthy of note that of the few aluminium yachts in existence three are due to German enterprise. The Aluminia, owned by the Prince Zu Wied, is a *bonâ fide* cruiser; the Susanne and the Luna were built for racing. The former is the property of her designer, Professor

Otzen, and the Luna was constructed to the order of Mr. B. Arons, from designs by Mr. Max Oertz. It cannot be said that the results have been satisfactory so far; but it is a healthy sign of the flourishing state of yacht architecture in Germany, that these costly experiments—for these yachts cost almost twice as much as similar craft of composite build—were carried out entirely in the interests of the science of naval construction.

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